POWER. and THE ENGINEER

EVERYTHING ABOUT THE GENERATION AND TRANSMISSION OF POWER

Vol. 32 NEW YORK, APRIL 5, 1910 No. 14

HOW much does it cost to run your power plant?

How much of that cost is paid to the man in responsible charge of it?

Do you pay him a salary—or wages?

Do you employ an engineer or hire an engine runner?

In an office building in a western city they paid the man in responsible charge of the heating, lighting, elevator service, water supply and the mechanical department in general ninety dollars a month, and let him live in the building. Lots of men are getting more than this for working short hours pasting bricks together with mortar or sticking tile into a mat of cement according to a picture which somebody has drawn for them, or even for cutting off pipe and screwing it together and wiping joints.

The time came when this building required an engineer, and one who refused to take it otherwise was engaged at $1200 per year and allowed to pay his fireman sixty instead of fifty dollars a month.

They were burning over ninety tons of coal a month—he has cut it to less than forty.

An increase in salary and wages of twenty dollars per month has resulted in a saving to the owners of this building of at least one hundred and twenty dollars per month in coal alone, and everything is spick and span. The machinery is kept up, and the repairs and renewals kept down by doing the things that ought to be done when they ought to be done and as they ought to be done.

This saving was effected not by an expensive and radical overhauling of the plant but by dangling that additional ten dollars per month in front of the fireman, by the introduction of new methods and the stopping of a hundred little wastes, and with such an absence of call upon the office for additional investment that it required an investigation by the auditor to prove that it was real.

There are hundreds of plants where savings as great as this can be made. The plant is very exceptional where the replacement of an ordinary mechanic by a real engineer will not save the difference in their cost and leave a handsome profit to the owner.

Which do you employ?

Step down into the engine room some day and find out. Spend a few minutes talking with him about his department, and see if he has the same intelligent grasp of its conditions and possibilities that you would require of the superintendent of another department of your business.

See if he has any ideas.

If he has, make his surroundings and pay commensurate with such a position, and not those of an irresponsible day laborer.

If he turns out to be nothing but a hand worker, put him on a laborer's job and get an engineer.

Plant of Pfister & Vogel Leather Co.

Modern Turbine Plant Providing for Electric Drive Will, It Is Estimated, Save $60,000 a Year in Operating Expenses over Old Corliss Engine Belted Units

BY OSBORN MONNETT

In this plant, which was designed and constructed under the superintendence of C. P. Bossert, chief engineer of the Pfister & Vogel Leather Company, Milwaukee, Wis., is found an interesting example of what can be done in the way of installing a maximum amount of power in limited space and at the same time providing the necessary auxiliary arrangements for economical and efficient operation. The case is one in which old-type belt drives with medium-power Corliss engines had from time to time been supplemented with belted generators for lighting and part of the power, until the facilities had become inadequate to take care of the increasing load in the many departments, and it became necessary to reorganize the power system, providing for electric drive throughout, having sufficient capacity for future growth. It is estimated that the change will show a saving of $60,000 a year in operating expenses.

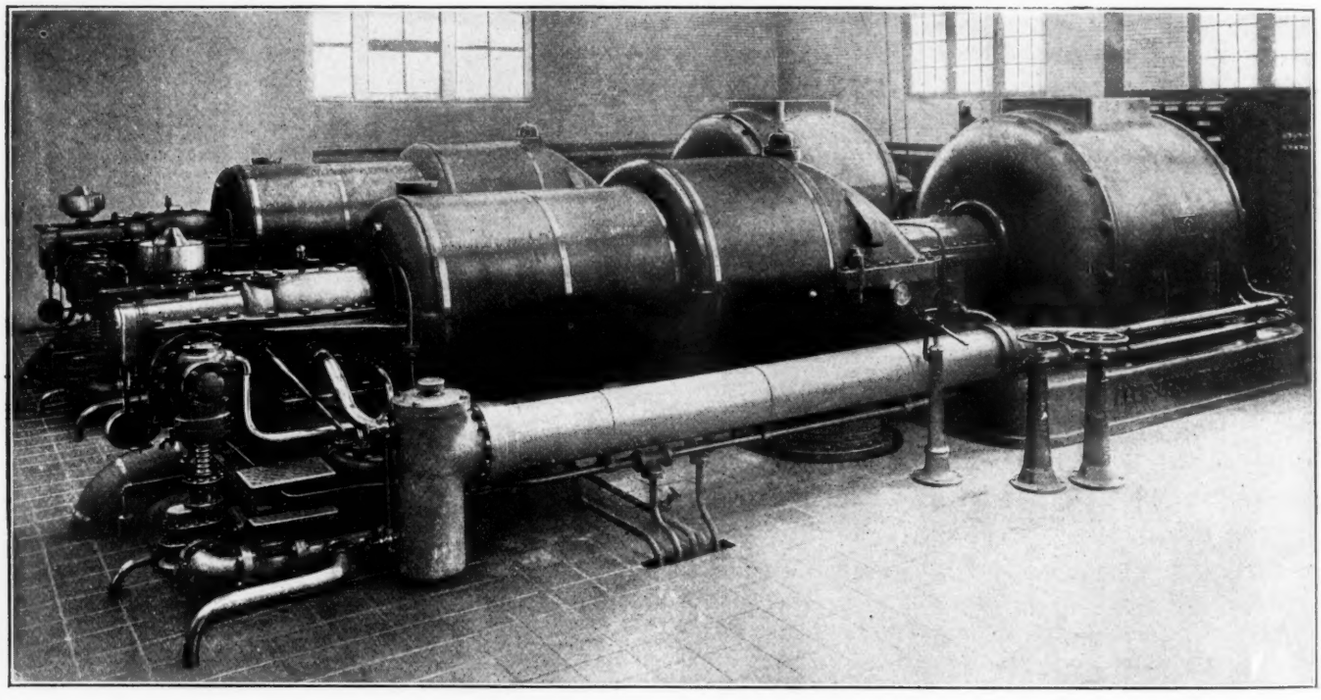


FIG. 1. TURBINE GENERATING UNITS IN NEW PFISTER & VOGEL PLANT

The new power plant is conveniently located to supply the various groups of buildings comprising the works and is close to the Menominee river, where condensing water may be easily obtained. Built of red pressed brick with stone trimmings and concrete foundation, the building occupies an area of 62x135 feet and is of the most substantial construction. Although saving of ground space was important, no restriction was placed on hight, and plenty of head room has been provided for light and ventilation. In the turbine room, which is spanned with a 10-ton traveling crane, the walls are lined to the ceiling with white enameled brick with a 5-foot green wainscoting around the bottom, while a substantial red-tile floor makes it easy to keep this part of the plant neat and clean. The stack is of Wiederholtz tile-concrete construction, and is well protected with lightning rods. It is 12 feet in internal diameter, 285 feet high, and occupies one corner of the turbine room. This is a result of the plan to economize floor space in every possible way. That portion of the stack inside the power house is heavily lagged with asbestos and painted white.

BOILER ROOM

Coal, consisting ordinarily of Youghiogheny screenings, is delivered by rail and dumped from the track level to a hopper in the basement. Passing, if necessary, through a motor-driven crusher, the coal enters the leg of a bucket conveyer, which takes it to an elevated steel hopper directly over the boilers, having a capacity of 800 tons. From this point a six-ton electrically operated traveling and weighing hopper distributes it to the stokers. Coal records for the station are therefore easily kept and by weighing before and after filling a furnace hopper, the coal consumption of any particular boiler may be put on record, using a system of blanks provided for that purpose. The Darbey system of vacuum ash handling has been installed, the ashes being raked into 8-inch pipes and elevated to an ash hopper of two carloads capacity, from which it is spouted to cars. The vacuum is maintained by a motor-driven exhauster, requiring 30 horsepower to operate.

Two rows of Edge Moor boilers containing three in each row will compose the complete steam-generating equipment, one row being now installed, while boilers from the old plant will be moved to the power house as soon as convenient. The boilers are of 585 horsepower rated capacity, having 5850 square feet of heating surface each, and are equipped with Green traveling link grates, 12 feet wide by 10 feet long, giving 120 square feet of grate surface. Foster superheaters are fitted in the setting, and superheat of about 175 degrees is carried, with a steam pressure of 160 pounds gage. Vulcan soot cleaners are installed on all the boilers to economize on labor, it being part of the

general plan to effect a considerable saving in this regard in comparison with the operation of the old plant. This feature has been brought to such a point that only one engineer, one fireman and one ash man are necessary for a shift. Two hours of the ash man's time are chargeable to removing ashes, twenty minutes to flue blowing, and the remainder of the time may be devoted to unloading coal from bottom-dump cars, distributing fuel to the hoppers and helping in the fire room.

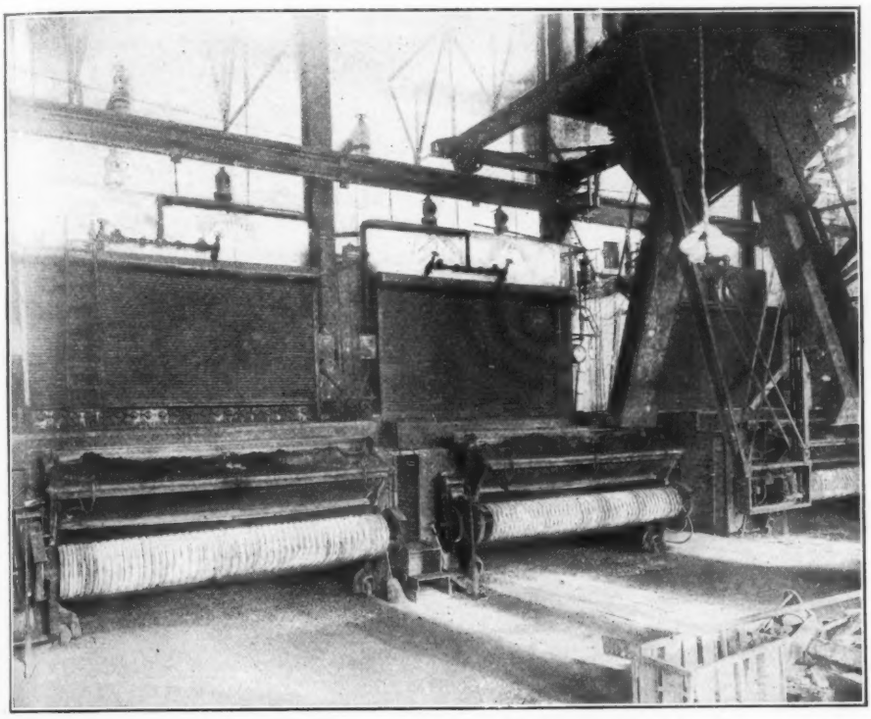


FIG. 2. BOILER ROOM, SHOWING TRAVELING COAL HOPPER

An ingenious utilization of space, made necessary by the circumstances, is found in the location of the receiving tanks handling condensation from the steam-heating mains throughout the plant. There are three of these tanks, of steel, embedded in the concrete foundation of the three north boilers, condensation from the heating mains being discharged into them by means of a vacuum pump. The corresponding space in the foundations for the opposite row of boilers is utilized for store-room purposes. Fairbanks tank pumps in duplicate discharge this water of condensation to a vertical open heater in the boiler room, which receives the exhaust of all steam-driven auxiliaries. When the water lowers to a predetermined point in the receiving tanks a fresh supply is automatically turned in from a water softener located on the property, or in an emergency, from the city mains. The feed water is handled by either of two 12 and 8 by 12-inch outside center-packed Fairbanks-Morse plunger pumps located at a convenient point on the boiler-room floor. These pumps are equipped with governors which control the feed-line pressure, but no automatic feed-water regulators are installed, although high- and low-water alarms are provided. Arrangements of the feed lines are exceptionally complete. Two branches of extra-heavy pipe extend under the floor in front of the boilers, cross-valved so that either branch may feed any or all boilers. In each branch is located a 5-inch Venturi feed-water meter, furnished by Builders Iron Foundry, of Providence, R. I., with the recording and indicating dials in the turbine room. With this arrangement it is possible to run a continuous evaporative test on the boiler plant as a whole, or any one boiler or battery of boilers may be tested, using one meter, while the thermometer takes care of the rest of the plant.

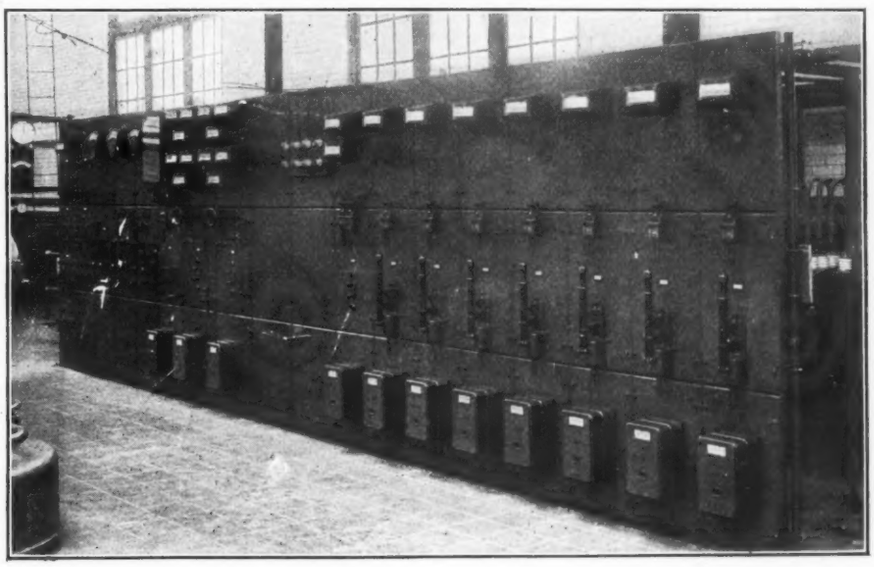


FIG. 3. REMOTE CONTROL SWITCHBOARD

The Venturi meters carry an indicating dial which shows the rate at which water is being used at any time, and in addition have a chart recorder from which is readily obtained the total quantity of water evaporated per 24 hours. In addition to this feature, recording steam and vacuum gages, draft gages, thermometers and Uehling CO_2 recorders are provided and constitute in their completeness a notable feature of the plant.

With the latter equipment samples of gas are taken from the uptake of each boiler by means of the suction generated by a jet of steam, purified, and sent to the apparatus in the turbine room where the gas is subjected to the action of a soda-ash solution, its resulting contraction operating the recording apparatus. Also connected to the instrument by small copper tubes are two direct-reading CO_2 gages for each boiler, one being located in the turbine room and one on the boiler front, convenient for inspection by the fireman. This elaborate equipment, in connection with the coal-weighing facilities, makes a combination which puts the operation of the plant on a strictly engineering basis.

As continuous operation is necessary, proper provision has been made against breakdowns of the piping system. The steam header forms a complete ring or loop of 8-inch pipe in a vertical plane behind the boilers, the lower side extending below the floor line. Valves in the header provide suitable means for making repairs without shutting down and nonreturn valves are provided at the boiler nozzles. Long-sweep bends drop from the steam line to large receiver separators in the basement, from which ex-

tend risers to the turbine throttles above. All lines of piping are properly drained with steam traps and the auxiliary lines carry superheated steam as well as the main headers.

Turbines and Condensers

For generating current, two 1500-kilowatt Allis-Chalmers turbo-generators are installed, delivering three-phase, 480-volt, 60-cycle current and running at 1800 revolutions per minute. Space is available for an additional unit.

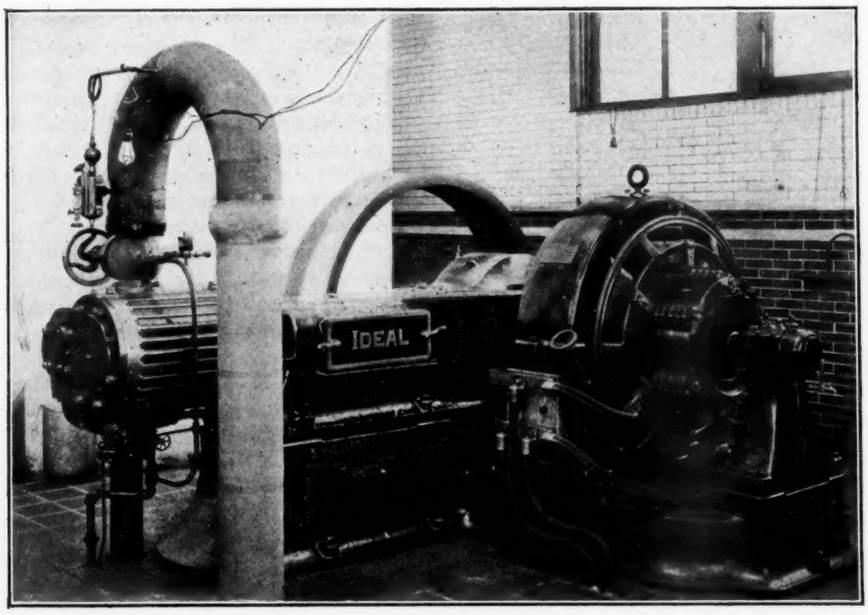


Fig. 4. One of the Steam-driven Exciters

In connection with the turbine a unique scheme has been devised for furnishing the steam for heating the buildings and carrying on the processes of manufacture. This consists in connecting a steam bleeder to an intermediate stage of the turbines and taking steam from it after it has accomplished a large amount of work. The pressure used in the heating and process steam lines is about five pounds gage, so the connection is made to the turbine at such a point that at half load the pressure is five pounds. At higher loads a greater pressure exists at this point and a reducing valve is provided. A nonreturn stop valve is placed on the line so that steam from one turbine cannot find its way to the other. Six thousand pounds of steam per hour may be taken from either one of the turbine casings without causing the opening of the auxiliary valve. As this steam contains no oil it is particularly adaptable for manufacturing purposes.

Two Tomlinson type-C condensers are installed, located directly beneath the turbines between the foundation piers. The selection of the proper sort of condenser was highly important, as the condensing water taken from the river is very dirty. The Menominee river, while fed from a creek, receives the discharge from numerous manufacturing establishments above the Pfister & Vogel plant, so that when the water reaches the inlet to the condensers it is in poor condition for condensing purposes. The condensers, which maintain a vacuum within 1.5 inch of absolute, with injection water at 60 degrees, are of somewhat novel design. The condenser head has a pear-shaped body provided with a spray nozzle for the water, so arranged that the openings are not restricted. This prevents any clogging due to the extremely dirty water. The pumping unit for each condenser consists of two single-stage centrifugal pumps of special design, mounted on the same shaft and driven by an 85-horsepower induction motor at 490 revolutions per minute. One pump takes water from the condenser head and discharges it into the hotwell against atmospheric pressure. The other takes water from the intake and discharges it through an air-ejector nozzle which is connected by suitable piping to the condenser head. This serves the purpose of the ordinary dry-air pump. The overflow from the auxiliary jet being cold, flows to the river through an independent connection, or if desired it may be diverted to the main injection supply to be used for condensing purposes, showing that no more water is necessary with this arrangement than with the ordinary jet condenser.

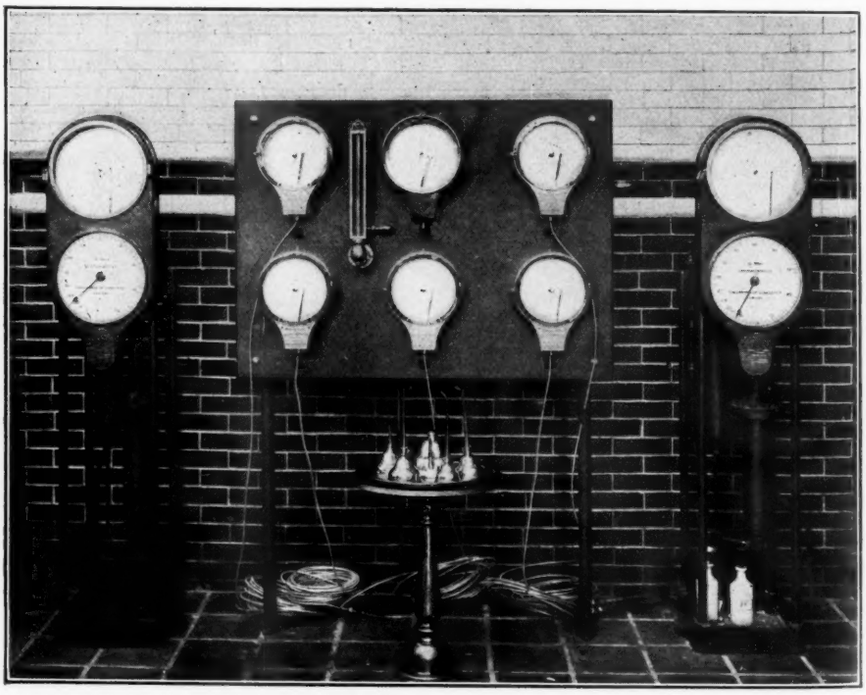


Fig. 5. Unusually Complete Equipment of Feed-water Meters and Recording Gages

Water Supply

A great deal of water is required in the process of tanning and it was found to be cheaper to purify the necessary supply than to buy it from the city. To accomplish this one of the largest water softeners ever built has been installed by the Kennicott company. This softener has a capacity of 150,000 gallons per hour and receives its supply from the hotwell of the condensers, in which the water is cleaner than that obtained from the river. This water is also hot, which assists in the process of softening. From the hotwell two De Laval turbine pumps, one steam-driven and one motor-driven, discharge the warm water into the softener as required. There is also held in reserve a Fairbanks-Morse tank pump for this purpose. Primarily this water

is used in manufacturing processes in the works, but incidentally it supplies water to the make-up tanks in the basement, as previously mentioned. After being pumped to the softener the water is not handled again by mechanical means, as it is at a sufficient hight to flow to the various departments by gravity. Buildings across the river are supplied with softened water through a 12-inch line laid in the bed of the stream. Two 1500-gallon Fairbanks-Morse underwriter's fire pumps in the basement of the power house complete the water system.

Electrical System

The output of the generators, governed by a voltage regulator, is distributed at 480 volts to induction motors for general power purposes in the factory. A dull-finished slate switchboard controls the distribution, the power panels being equipped with General Electric remote-control oil switches, time-limit relays, and Thompson polyphase recording wattmeters. Lighting circuits are operated through three 75-kilowatt transformers, at 110 volts, and by means of a double set of busbars on the board may be thrown also on the direct-current exciter circuits. Two steam-driven exciters are provided, either of which is large enough to excite both alternators. One machine consists of a 75-kilowatt Allis-Chalmers generator, directly connected with an Ideal engine, and the other is a 150-kilowatt De Laval turbine outfit. This latter can carry both exciting and lighting loads. All electrically driven auxiliaries in the power house operate with induction motors, with the exception of the crane motor in the turbine room, which is connected with the exciter circuits, to be able to make repairs when the main units are not running. A small Allis-Chalmers motor-generator set will be installed to furnish exciter current in ordinary operation.

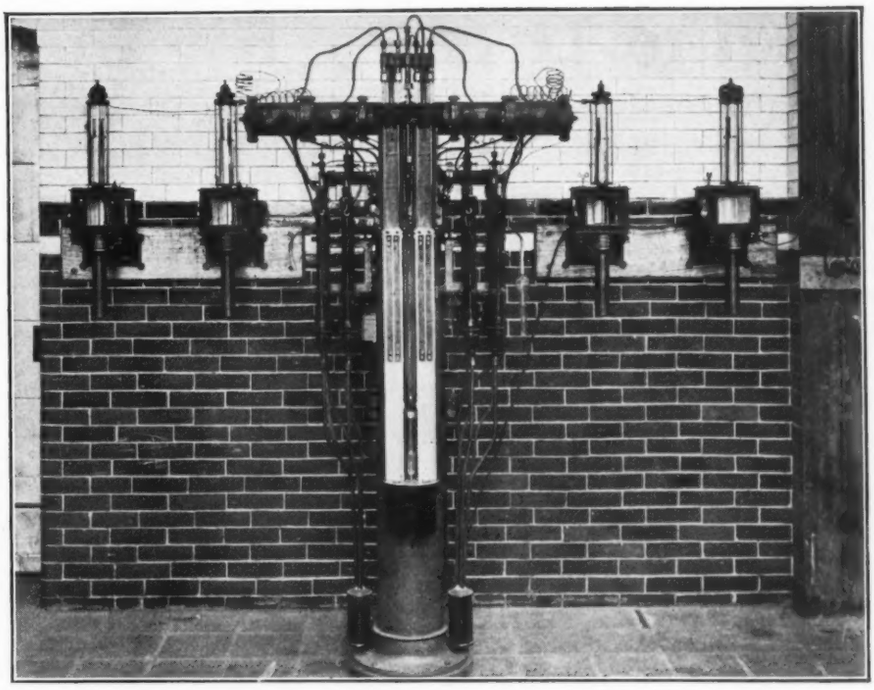


Fig. 6. Uehling Composometer for CO_2 Records

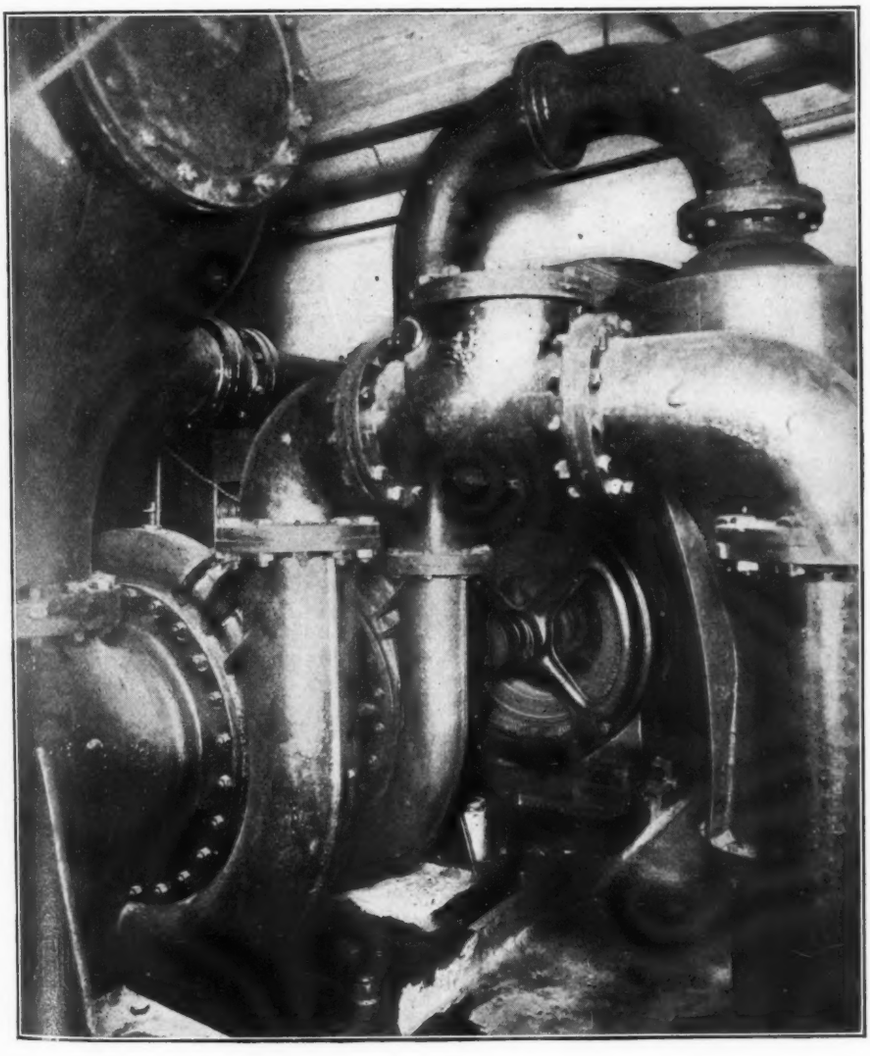


Fig. 7. Main and Auxiliary Condenser Pumps Mounted on Same Shaft

---

The total heat of saturated steam can be calculated fairly closely by a formula, based on Doctor Davis's experimental work, which Prof. R. H. Smith has recently proposed in *Engineering*. This formula is

$$H = 1826 + t - \frac{1{,}250{,}000}{(1620 - t)}$$

where H is the total heat and t is the temperature.

The School of Mines of North Dakota is installing a briquetting plant for lignite, with a capacity of two tons per hour and a complete gas plant of the latest type with automatic recording apparatus for temperatures, pressures, etc., together with a most complete laboratory for coal and gas analysis and testing.

Engine Driving by Compressed Air

Current from Long-distance Transmission Line Drives Motor-operated Compressor Supplying Engine with Air Which Is Reheated on Way to Cylinder. Economy of System

BY SNOWDEN B. REDFIELD

Compressed air as a medium for driving reciprocating devices such as rock drills, pneumatic tools and even steam pumps is well established in the field of engineering. Under certain conditions compressed air is sometimes used for the driving of reciprocating engines with flywheels, valve gear and all the appurtenances of a regular steam unit. Most compressed-air installations, however, are anything but economical from a fuel standpoint and indeed in the case of rock drills and pneumatic tools the principal object is to get the work done with economy of labor rather than with economy of fuel.

In many cases it would be quite convenient to drive steam engines by compressed air and this is being done at the shafts of some of the Western mines, for operating hoists of large power. Up to the present time electricity has not been successfully applied in this country to hoists of great depth and in cases where the power is obtained from long distances by electric transmission it is very convenient to transform this electricity into compressed-air power for the driving of the hoisting engines. In these cases economy of power is much to be desired even if that power comes from a waterfall, for the more economically this power is used the greater percentage of the possible output of the plant, transmission wires, etc., representing a given amount of money invested, can be usefully employed.

Owing to these developments there have been at various times, inquiries as to the amount of air necessary to operate engines of given size, and the economy that can be obtained thereby or, in other words, the amount of fuel to be consumed to develop a given horsepower in the engines which are operated by the compressed air. Very few actual data from experiments are available on these subjects, but the figured results shown by the accompanying diagrams give much information which should be of value in this connection. These results have been figured not only for theoretical conditions, but also for what is more important; practical conditions as well. The curves plotted from these figures will then give the results which should be obtained with perfect apparatus and also with ordinary machinery in commercial use. The theoretical results will be found useful for studying the subject and for the checking up of actual results obtained from the machines themselves. The figures for practical performance are based upon actual conditions of operation such as would be obtained in any well organized plant, taking into consideration every probable defect in the mechanism and operation, which it would not be possible to avoid. Results obtained from these curves based upon what should actually occur in a real plant should be found especially valuable for estimating purposes, for the installation of new equipments.

The Proposition

It is proposed in the calculations from which the curves are plotted to drive a reciprocating steam engine having a flywheel and valve gear, the latter being capable of suitable adjustment to give the results desired. Compressed air is supposed to be furnished by an ordinary reciprocating compressor of modern design and ordinary performance, this compressor for the purposes of fuel calculation being supposed to operate by steam.

It is well appreciated that to obtain economy in an engine of any sort expansion of the gas operating in the cylinder is necessary. In other words, there must be cut-off in the engine cylinder. Expansion of compressed air, as is well known, when operating behind a working piston, leads to excessive drop in the temperature of the air. This drop in the temperature simply indicates that the energy in the air, which is, of course, the heat in the air, is being given up and transformed into work which is being delivered to the shaft of the engine. Expansion of compressed air in an engine cylinder is not greatly different from adiabatic expansion because there is very little chance for heat to be given to the air inside the cylinder by radiation inward from the cylinder walls. Consequently the temperature of the air at any point in the expansion curve can be calculated within a very few degrees by the ordinary adiabatic formula.

If expansion in the engine cylinder using air is carried to too great an extent the temperature of the air in the cylinder will fall below 32 degrees Fahrenheit and consequently the moisture in the air will freeze, this freezing leading to the clogging up of the exhaust ports of the engine, the formation of ice all over the outside parts of the cylinder and very probably the stopping of the engine altogether. For this reason it is essential in using compressed air in an engine cylinder to see to it that the temperature of the expanding air does not fall below the freezing point.

In all the calculations connected with this case it is assumed that the terminal temperature of the expanded air in the engine cylinder shall fall to 35 degrees Fahrenheit and no lower. Under this assumption the engine is supposed to be given an earlier and earlier cutoff to give greater and greater economy. Earlier cutoff will, of course, mean greater ratio of expansion and this in turn means a greater drop in temperature of the air. In order, therefore, to insure that the terminal temperature of the expanded air shall not be lower than 35 degrees, it is assumed that the air entering the engine cylinder is heated to a temperature such that with the given cutoff, the final temperature shall not be below this point.

The Theory First

To study the conditions imposed it will be well to look first at the theoretical side of the question. Glancing at the accompanying chart, in the upper left-hand portion will be found four indicator diagrams. The two upper ones are for strictly theoretical conditions, the left-hand diagram being that of the air cylinder of the compressor and the right-hand diagram that of the working cylinder of the engine being driven by the compressed air. The two indicator diagrams below this represent practical conditions, taking into account all the usual losses due to imperfect mechanical devices and faults in operation.

To make the theoretical case simpler it is assumed that the temperature of the atmosphere is 35 degrees Fahrenheit. The air at this temperature enters the cylinder of the compressor and in a single stage is compressed up to 90 pounds gage pressure, when as shown by the upper left-hand diagram its temperature will be 415 degrees. This air is then supposed to enter the engine cylinder without any loss of pressure or temperature. If now the valve gear of the engine is set to cut off at just about 0.26 the air in this cylinder will be expanded down to atmospheric pressure and its temperature will have been reduced to 35 degrees. At this temperature and atmospheric pressure it is assumed that the air is exhausted to the atmosphere.

Such an arrangement as this would mean ideal conditions and the efficiency calculated from the ratio of the work done in the engine cylinder to that done in the air-compressing cylinder will be 100 per cent. Furthermore, the mean effective pressure theoretically developed in the engine cylinder would be about 39 pounds and the indicated horsepower

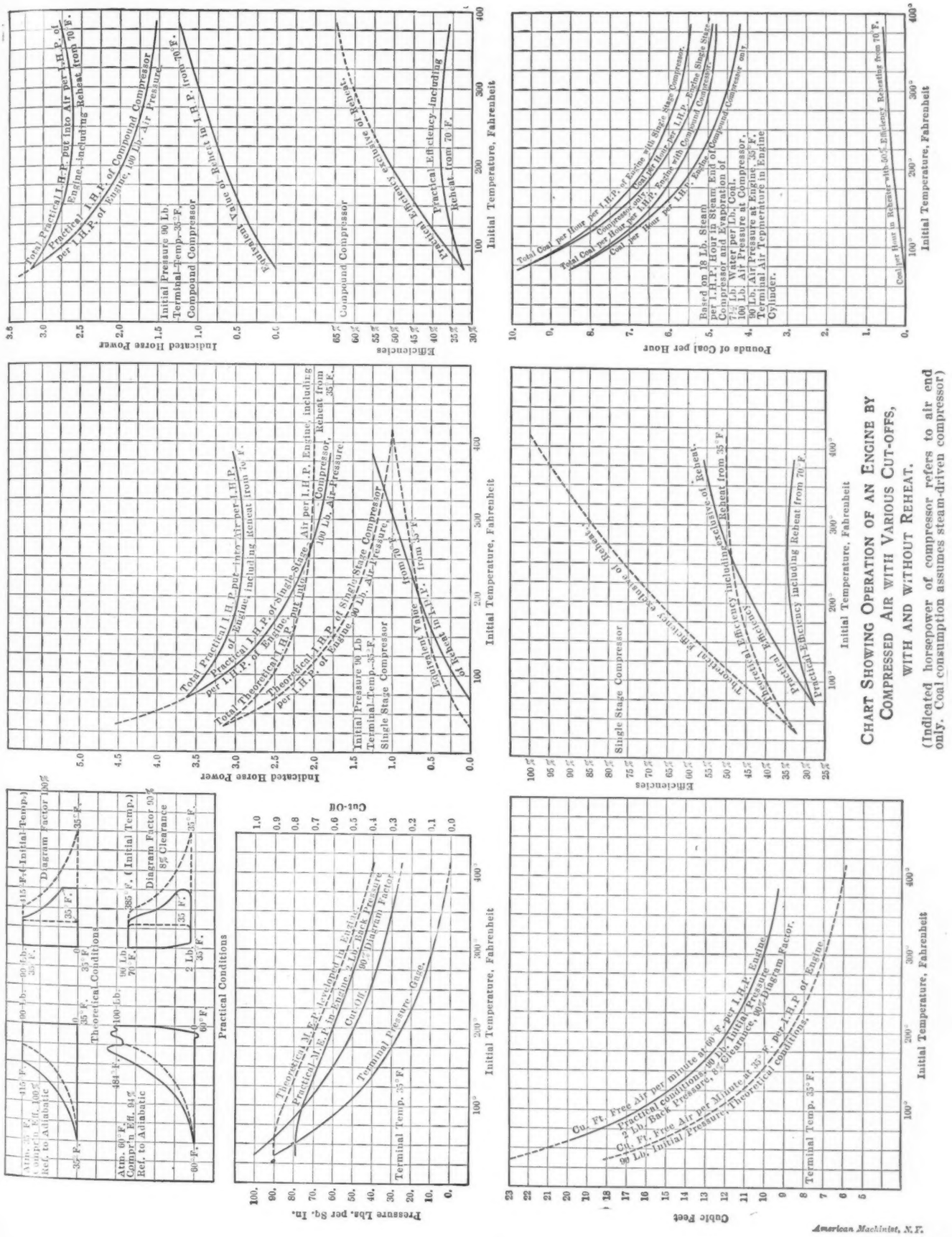

Total Practical I.H.P. put into Air per I.H.P. of Engine, including Reheat from 70° F.
Practical I.H.P. of Compound Compressor per I.H.P. of Engine, 100 Lbs. Air Pressure.
Initial Pressure 90 Lb.
Terminal Temp.-35° F.
Compound Compressor
Saving due to Reheat in I.H.P. from 70° F.
Theoretical Efficiency including Reheat from 70° F.
Practical Efficiency including Reheat from 70° F.
Compound Compressor
Indicated Horse Power
Efficiencies
Initial Temperature, Fahrenheit
Total Practical I.H.P. put into Air per I.H.P. of Engine, including Reheat from 70° F.
Practical I.H.P. of Single Stage Air per I.H.P. of Engine, 100 Lb. Air Pressure.
Total Theoretical I.H.P. put into Air per I.H.P. of Engine.
Theoretical I.H.P. of Single Stage Compressor per I.H.P. of Engine, 90 Lb. Air Pressure.
Initial Pressure 90 Lb.
Terminal Temp.-35° F.
Single Stage Compressor
Saving due to Reheat from 70° F.
Initial Value of Reheat from 35° F.
Theoretical Efficiency including Reheat from 35° F.
Practical Efficiency including Reheat from 35° F.
Single Stage Compressor
Total Coal per Hour per I.H.P. of Engine with Single Stage Compressor.
Total Coal per Hour per I.H.P. Engine with Compound Compressor.
Coal per Hour per I.H.P. Engine with Compound Compressor only.
Based on 18 Lb. Steam per I.H.P. Hour in Steam End of Compressor and Evaporation of 7½ Lb. Water per Lb. Coal.
100 Lb. Air Pressure at Compressor, 90 Lb. Air Pressure at Engine. 35° F. Terminal Air Temperature in Engine Cylinder.
Coal per Hour in Reheater with 50% Efficiency Reheating from 70° F.
Pounds of Coal per Hour
Initial Temperature, Fahrenheit
Atm. 35° F. 415° F. Compr. Eff. 100% Ref. to Adiabatic
90 Lb. 35° F.
Diagram Factor 100%
Theoretical Conditions
Atm. 60° F. Compr. Eff. 94% Ref. to Adiabatic
100 Lb. 484° F.
90 Lb. 70° F.
385° F. (Initial Temp.)
Diagram Factor 93%
8% Clearance
2 Lb.
60° F.
Practical Conditions
Cut-Off.
Theoretical M.E.P. developed in Engine.
Practical M.E.P. in Engine.
2 Lb. Back Pressure.
90% Diagram Factor.
Terminal Temp. 35° F.
Terminal Pressure—Gauge.
Pressure Lbs. per Sq. In.
Initial Temperature, Fahrenheit
Cu. Ft. Free Air per minute at 60° F. per I.H.P. Engine
Practical conditions. 90 Lb. Initial Pressure
2 Lb. Back Pressure, 8% Clearance, 90% Diagram Factor.
Cu. Ft. Free Air per Minute at 35° F. per I.H.P. of Engine.
90 Lb. Initial Pressure. Theoretical conditions.
Terminal Temp. 35° F.
Cubic Feet
Initial Temperature, Fahrenheit

CHART SHOWING OPERATION OF AN ENGINE BY COMPRESSED AIR WITH VARIOUS CUT-OFFS, WITH AND WITHOUT REHEAT.

(Indicated horsepower of compressor refers to air end only. Coal consumption assumes steam-driven compressor)

American Machinist, N.Y.

developed in the air-compressing cylinder per indicated horsepower developed in the cylinder of the engine using the compressed air would, of course, be 1.

This is the extreme and only theoretically possible case. The worst possible case that we could have under theoretical conditions would be to assume that after the air has been discharged from the compressing cylinder at 415 degrees and 90 pounds pressure it should have become entirely cooled down while passing through the air pipe leading from the compressor to the engine, so that its temperature when reaching the latter is only 35 degrees. The pressure is assumed to remain at 90 pounds. Under these circumstances it is, of course, impossible to allow any expansion of the air at all because no drop in temperature is allowable or freezing will take place. The engine must then admit air through the full stroke, when the exhaust valve would open and the air would expand simply into the atmosphere without doing any useful work and without any appreciable temperature drop.

Under these circumstances the indicator diagram would be represented by a rectangle, there being no slanting expansion curve. This would, of course, give very poor economy because although theoretically 90 pounds mean effective pressure is developed in the engine cylinder, there being no expansion a very large volume of air must be used, comparatively speaking, in order to do a given amount of work. To develop 1 indicated horsepower in the engine cylinder under such conditions, without expansion, it will be found that the compressor will be called upon to develop about 3.2 horsepower in its air-compressing cylinder, giving a theoretical efficiency of only about 32 per cent.

Between these two cases of the theoretically perfect, that is, admitting air at 415 degrees directly from the compressor, expanding down to atmosphere and 35 degrees, and the case where the air is admitted to the engine cylinder at 35 degrees and no expansion at all is allowed, there is a wide range of possible conditions. In the calculations, therefore, it is assumed that the air coming from the compressor is admitted to the engine cylinder at various temperatures ranging all the way from 35 degrees as a minimum, to 415 degrees as a maximum and in each case the cutoff is so set that the air in the engine cylinder will expand down until at the end of the stroke, its temperature has dropped to 35 degrees. Such an intermediate indicator diagram is shown by the full lines in the upper left-hand section of the accompanying chart. Such conditions will, of course, give rise to an economy which is intermediate between that of the perfect theoretical conditions and the most imperfect theoretical conditions, as described.

Cutoff, Mean Effective and Terminal Pressures

To follow through the results which would be obtained with various initial temperatures of the air entering the engine cylinder it will be necessary to examine the various diagrams on the chart. Looking first at the middle diagram on the left-hand side of the chart, we will see that when the air enters the engine cylinder with an initial temperature of only 35 degrees, the cutoff, as shown by the line so marked, will be unity, that is, there will be no cutoff at all. This, as shown, will result in a mean effective pressure in the engine cylinder of 90 pounds, there being no loss assumed under these theoretical conditions. There being no expansion the terminal pressure will also be 90 pounds as shown by the curve marked "Terminal Pressure."

Following these curves a little further we will find that if the air is admitted to the engine cylinder at 100 degrees Fahrenheit it will be possible to so set the engine valves that the cutoff may occur at about 0.75 of the stroke. This will result in a theoretical mean effective pressure developed in the engine cylinder of 85 pounds and the terminal pressure by the time the temperature has dropped to 35 degrees will be found to be 53 pounds. At 300 degrees, under these theoretical conditions the cutoff may be set at 0.35 stroke. The theoretical mean effective pressure developed in the engine cylinder will then be about 53 pounds and the terminal pressure will have fallen to about 9 pounds, the terminal temperature, of course, being 35 degrees as in every case. If the air is allowed to enter the engine cylinder at 415 degrees, the results will be as already outlined for the most perfect theoretical conditions.

Quantity of Air Required

After this study of the engine indicator diagram we may determine the quantity of free air required per minute to develop one indicated horsepower in the engine cylinder, under these same varying conditions. Looking first at the theoretical state of affairs as shown by the dotted curve in the lower left-hand section of the chart we will find that with 35 degrees initial temperature in the engine cylinder it will require a little over 18 cubic feet of free air per minute to develop one indicated horsepower. At 300 degrees initial temperature about 7 cubic feet of free air and at 415 degrees about 5 9 cubic feet, will be required per minute to develop one indicated horsepower in the engine cylinder. In each case this air is measured at atmospheric pressure and 35 degrees temperature.

Reheating

At this point something should be said about reheating, as this is one of the most important points in the use of air in engines having cutoff. So far we have assumed that the air delivered to the engine has simply cooled off to a certain point as it passes through the pipe line between the compressor and the engine. Under actual conditions it usually happens that the air has cooled down to very nearly atmospheric temperature before reaching the engine and consequently in order to obtain economy by the use of cutoff in the engine it is necessary to reheat the air in some suitable device known as a reheater, before allowing it to pass into the engine cylinder. This reheating of the air, of course, means the consumption of additional fuel but owing to the nature of compressed air itself, this additional heat put into the air will develop power in the engine cylinder with an economy practically unattainable by any other method of transforming heat energy into motion. By means of separate reheaters, therefore, it is possible not only to use compressed air in an engine where conditions make it imperative to use air instead of steam, but it is also possible to develop power with a very reasonable degree of economy from the coal pile.

When burning coal under a steam boiler an economy of 70 or 75 per cent. is possible and often attained. With a properly designed reheater for heating the compressed air before entering the engine, it is undoubtedly possible to obtain an economy equal to this but under ordinary working conditions the reheater in an air plant receives but scant attention unless a special man is hired for this purpose. Indeed, considering the neglect which reheaters receive under usual conditions the economy obtained by their use is wonderful. However, not to favor the reheater too much it is assumed in these calculations that the efficiency of the transference of the heat from the coal to the air is only 50 per cent. In other words, only one-half of the heat available in the coal is assumed to have been put into the compressed air in the reheater before actually entering the inside of the engine cylinder.

This compares with rather poor steam-boiler practice but at this point the analogy ceases for with the steam plant although the steam may contain 70 or 75 per cent. of the heat of the coal, only about 12 or 15 per cent. of this heat is actually transformed into work in the engine cylinder. On the contrary, when using heated air in an engine cylinder the economy of the transformation of the heat in the air into work may well reach 80 per cent.

Power in the Reheater

All heat which is put into the air in the reheater must, of course, represent additional energy from the coal pile just the same as the energy which is put into the air by its compression in the air compressor and therefore the coal burned in the reheater should be charged against

the plant in obtaining the total economy. In the upper central diagram and also in the upper right-hand diagram showing power curves will be found curves marked "equivalent value of reheat in indicated horsepower." The dotted curve, representing the theoretical conditions, assumes that the air is reheated from the temperature of the atmosphere, namely 35 degrees Fahrenheit, to the various initial temperatures marked underneath the curves. Practical conditions will be described further on.

Examining the upper central diagram it will be found that for the theoretical conditions as shown by the dotted line the equivalent value of the reheat is, of course, zero, when the air enters the engine cylinder at 35 degrees and when the initial temperature is raised to 100 degrees it is seen that the heat in the air is equivalent to about 0.37 indicated horsepower per indicated horsepower developed in the engine cylinder by the hot compressed air.

This increases as the temperature is raised until finally, when the air is reheated to an initial temperature of 415 degrees, corresponding to the temperature at which it left the compressor cylinder, the heat put into the air is equivalent to one indicated horsepower per indicated horsepower developed in the engine cylinder.

We have already found that in order to compress the air up to 90 pounds pressure and develop one indicated horsepower in the engine cylinder under the theoretically perfect conditions, one indicated horsepower has also been developed in the compressing cylinder. This goes to show, therefore, that all of the work which is done in the engine cylinder has come from the heat in the air and as it has been assumed that the compressed air has entirely cooled down to 35 degrees and then been reheated by the separately fired reheater to 415 degrees, it is this extra heat put into the air in the reheater which does the work in the engine cylinder. As far as the work done in the compressor is concerned this latter is all thrown away when the air in the pipe line has been allowed to cool down to its original atmospheric temperature. In other words, under these circumstances, the compressor has done one indicated horsepower of work, which has then been lost in radiation and the reheater has added another indicated horsepower in the shape of heat, making two indicated horsepower expended upon the air to develop one indicated horsepower in the engine cylinder. This is a theoretical efficiency of 50 per cent.

Efficiencies

All of these factors of horsepower developed in the compressor; horsepower put into the air in the reheater, and total indicated horsepower put into the air per indicated horsepower in the engine, will be found in the upper central diagram. In the lower central diagram will be found the corresponding curves for efficiency. Two of these curves are marked "efficiency exclusive of reheat." By this is meant that no reheater is used, but that under the conditions of operation the air from the compressor has simply cooled down by radiation in the discharge pipe, from its natural temperature of 415 degrees, to the various initial temperatures shown at the bottom of the diagram and that the work done has actually come from the compressor only. This work of the compressor is represented by the line in the diagram just above, marked "indicated horsepower of single stage compressor per indicated horsepower of engine," etc. The indicated horsepower of the compressor refers to the air end only, in all cases.

Practical Conditions

So much for theoretical conditions. Theoretical conditions are all right to study the situation and determine the method of calculation, but practical results are what are desired. Looking now at the two indicator diagrams marked "Practical Conditions" in the upper left-hand corner of the chart, it will be found that the following practical considerations are assumed: The atmosphere is assumed to be 60 degrees Fahrenheit, and the compression efficiency of the air cylinder of the compressor is assumed to be 94 per cent., that is, 6 per cent. loss over and above theoretical adiabatic conditions; 88 per cent. mechanical efficiency of the compressor is also allowed. Further, it is assumed that there is a ten-pound pressure drop in the pipe line from the compressor to the engine and consequently the compressor must pump the air to 100 pounds pressure, in order to reach the engine at 90 pounds. Also it is assumed that under the worst case of cooling of the air from the compressor, the initial temperature of the air in the engine cylinder is 70 degrees, or 10 degrees above the atmosphere. It is from this temperature that all the reheating is calculated under the practical conditions. Furthermore, the back pressure or exhaust of the engine using the compressed air, is considered to be two pounds above the atmosphere.

To allow for the rounding of the corners of the diagram, due to the operation of the valve gear, a diagram factor of 90 per cent. is used; or the mean effective pressure actually developed in the engine will be 90 per cent. of what would theoretically be obtained under the conditions of initial pressure and the given cutoff. As a final practical condition, the engine cylinder is supposed to contain 8 per cent. clearance, this clearance operating to increase the amount of air required. Just as in the theoretical condition, it is likewise assumed that the final temperature of the air after expansion is 35 degrees Fahrenheit, or just above freezing. No leakage in the system is taken into account, as it is considered that the pipe lines and connections for a permanent installation would have no more leaks than a similar steam pipe line leading from a boiler to an engine.

Practical Temperature Range

Under the best practical conditions allowing the air in the engine cylinder to expand down to two pounds above atmosphere for the back pressure, the initial temperature in the cylinder can be 385 degrees Fahrenheit. With these conditions a cutoff of about 0.27 can be used, resulting in a practical mean effective pressure in the engine cylinder, with all the above enumerated losses taken into account, of about 37 pounds. These conditions are shown by the dotted indicator diagram in the lower set, having these temperatures and pressures marked upon it, in the upper left-hand portion of the chart.

From the chart it will be seen that 385 degrees Fahrenheit is the maximum temperature of reheating which is practically considered. Of course, it would be perfectly possible to heat this air to a much higher temperature provided lubrication troubles were not encountered, but under these circumstances one of two things would happen. First, if the temperature were greater than 385 degrees, and the expansion were carried out to 35 degrees, the terminal pressure of the air in the cylinder would be below the back pressure; producing a loop in the indicator diagram. If, however, with an initial temperature greater than 385 degrees Fahrenheit, the expansion were carried down just to meet the back pressure line without producing either a drop or a loop, the temperature of the air in the cylinder at that point would be considerably above 35 degrees, and consequently we would not have obtained from the air all of the work available in the heat. This, therefore, would result in loss of economy. For these reasons the calculations have been stopped, as shown, with an initial temperature of 385 degrees Fahrenheit as a maximum.

Glancing at the full line curves in the lower left-hand diagram of the chart, this curve being marked "cubic feet free air per minute at 60 degrees Fahrenheit per indicated horsepower engine, practical conditions, etc.," it will be found that with a minimum initial temperature of 70 degrees Fahrenheit, about 20 cubic feet of free air per minute per indicated horsepower developed in the engine will be required. This falls off as shown by the curve until, with the most advantageous practical conditions of 385 degrees initial temperature, about 9.3 cubic feet of free air will be required per minute per indicated horsepower of the engine.

Power Required

In the upper central diagram will be found the full line curves of practical conditions, showing the horsepower developed in the compressor, that added in the reheater and the total horsepower put into the air per indicated horsepower developed in the engine cylinder. In the lower central diagram will be found also the full line curves of practical efficiency.

All of the curves so far considered assume the use of a single-stage compressor. Glancing at the upper right-hand set of curves on the chart, there will be found results of practical considerations only and this time assuming the use of a compound compressor. All of the losses, as already outlined for the practical case, are included.

Maximum Power Economy

From the two curves of practical efficiency, including reheat from 70 degrees, this efficiency being from indicated horsepower in the engine cylinder to the indicated horsepower in the compressing cylinder of the compressor, it will be noted that a point of maximum efficiency is reached and so it would appear that there would be no advantage in reheating the air beyond 275 degrees when using a single stage compressor, or 250 degrees with a compound compressor. The curves show a decrease in efficiency after these temperatures are exceeded. As far as efficiency is concerned, this is true, but this is not the whole story. The efficiency in these particular cases does not mean dollars and cents directly, for, as already pointed out, the heat put into the air in the reheaters is developed into work in the engine cylinder at an extremely economical rate.

In order to understand this, it is necessary to glance at the lower right-hand set of curves as shown in the chart. Here the calculations are referred back to the coal pile, and all the results are strictly practical, the conditions imposed being as shown on this diagram. These conditions assume 18 pounds of steam per indicated horsepower per hour in the steam end of the compressor and an evaporation of 7½ pounds of water per pound of coal in the boiler, in addition to all the other practical conditions as to compression, expansion, pressures, temperatures and clearance already enumerated.

Coal Pile Economy

In this set of curves it will be found that there is one marked, "coal per hour per indicated horsepower of the engine, single-stage compressor only," and another one marked, "coal per hour per indicated horsepower of the engine, compound compressor only." These curves, as may be assumed from their marking, do not include the coal put into the reheaters, and may, therefore, be taken as the results, assuming that the air from the compressor has simply cooled down in its passage from the compressor to the engine, to the various initial temperatures as marked at the bottom of the curve. At the extreme bottom of the diagram will be found another curve which is marked, "coal per hour in the reheater with 50 per cent. efficiency, reheating from 70 degrees Fahrenheit." As intimated, this curve assumes that only one-half of the heat available in the coal burned in the reheater is actually present in the air after passing to the engine.

All the results in these curves are per indicated horsepower developed in the engine, and it will be found that with 385 degrees initial temperature, the compound compressor will require about 4.2 pounds of coal per hour, and the single-stage compressor will require about 4.9 pounds of coal per hour. However, glancing at the curve for the reheater, it will be found that at 385 degrees Fahrenheit only a little over one-half pound of coal per hour is required to heat the air to this temperature. The quantity of coal required in the reheater as given by the lower curve, is added to each of the upper curves marked, for the "compressor only," so as to give the total coal per hour per indicated horsepower of the engine, both with a single-stage and a compound compressor, as shown. This total quantity, it will be seen varies from 9.8 pounds per hour with no reheat and a single-stage compressor, to 5.4 pounds with reheating to 385 degrees. For the compound compressor this total coal per hour varies from 8½ pounds to 4.8 pounds per hour per indicated horsepower of the engine, according as the reheat increases and the cutoff shortens, giving greater economy.

It will be noticed from these latter curves that a point of minimum coal used is not quite reached as far as the calculations are carried out. It has already been explained why a higher temperature of reheating has not been figured upon. This then shows that reheating is economical from the dollars and cents point of view, even beyond the point where economy seems to decrease from the strictly B.t.u. point of view. This is due simply to the fact, as already explained, that the B.t.u. put into the air in the reheater are put in at a much cheaper rate than those put into the compressor through the steam cylinders of the latter. As a matter of fact, the trend of the curve for total coal per hour in the compound compressor, including reheat, does appear to reach just about a minimum point between 385 and 400 degrees Fahrenheit. This point, therefore, would seem to be the maximum reheat temperature for practical conditions. In fact with initial temperatures greater than this, trouble will be found to come from the carbonizing of the lubricant, so this again puts a practical limit at this point.

Cooling Surface in Condensers

R. M. Neilson in a recent paper upon surface condensers says:

The weight of steam which can be condensed per square foot of cooling surface varies enormously according to the conditions.

In reciprocating-engined steamships it is usual to allow from 1 to 1½ square feet of cooling surface per indicated horsepower. Allowing for water carried over from the boilers and condensation of steam in the engine, etc., we may take the mean condensation rate in the condenser as from about 10 to 14 pounds of steam per square foot. The vacuum is rarely above 26 inches.

In turbine power stations it is usual to allow from 3 to 4½ square feet of condenser cooling surface per kilowatt of rated capacity of the turbines, so that the condensation rate is from three to six pounds per square foot at the full-rated load of the turbines. The turbines are, however, usually expected to carry a considerable overload.

The condensation rates which can be obtained under suitable conditions are greatly in excess of these figures as shown by the following references.

Professor Weighton with an experimental contraflo condenser has dealt with 35 pounds of steam and water exhausting from a reciprocating engine per square foot of cooling surface per hour, and maintained a vacuum of 28 inches. (*Proceedings* of Institution of Naval Architects, April, 1906.) With lower vacua, condensation rates greatly in excess of this have been obtained. The author with a small contraflo condenser working at atmospheric pressure has condensed nearly 80 pounds of steam per square foot per hour with no evidence that a limit had been reached. Both Perry and Holmes state that Joule obtained a condensation rate of 100 pounds of steam per square foot, the pressure not being given. ("The Steam Engine," Perry, first edition, page 424. "The Steam Engine," Holmes, fourth edition, page 428.)

Flue Collapses in Lancashire Boiler

Accident to English Boiler Causes Death of Fireman and Extensive Damage to Property. Expansion of Long Flue and Possibly Scale Suggested as Cause of Disaster

BY JOHN S. LEESE

A serious boiler explosion, resulting in the death of one man and minor injuries to a number of other persons, besides causing extensive damage to property recently occurred at a cotton mill in England. The boiler, of the two-flue Lancashire type, 8 feet 6 inches diameter by 30 feet long, with flues 3 feet 2 inches diameter, made by Fowler & Sons, was the middle one of a battery of three, situated in a basement on the west side of the mill. The working pressure was between 150 and 160 pounds per square inch, the safety valves being set to blow at 170 pounds, which was the pressure authorized by the insurance company. The boiler was installed in January, 1904, and had not been tested since installation. Internal inspection by the insurance company took place annually and the last occasion was in August, 1909. The boiler was externally inspected every three months, January being the last month of inspection. Cleaning took place every month or five weeks, the last time being the two days before the explosion, which occurred on January 24. The fires were lighted on Sunday evening, January 23, and banked over night, the boiler being put into commission at 6 a.m. on Monday morning. A new mechanical stoker was put into operation for the first time on the morning of the accident. The other boilers have recently had mechanical stokers added and have naturally been working at a rather higher rate of steam production than before. At 11 o'clock the accident occurred. One of the firemen on duty was unhappily so badly scalded that he died the next morning. He was, however, able to make the statement that the water was within 3 inches of the top of the glass about five minutes before the flue let go. In the evidence at the coroner's inquest it was stated that the pressure gage, valves, water gages and low-water alarm were in good order. All three boilers were connected into the same steam main and all safety valves were set to blow at the same point, so that the theory of excessive pressure cannot be advanced.

The fireman who escaped injury gave evidence that at 11:10 a.m., the deceased and himself were in the boiler house when he saw flames issuing from the furnace at the front of the boiler. Thinking that they were caused by the forced-draft steam blast he shut it off. He then heard a hissing noise in the furnace louder than the noise of the blast. This noise struck him as being an escape of water or steam from the boiler into the furnace. He then called to the deceased and made for safety. The deceased met with his injuries while trying to cut the boiler out of the steam main to prevent the other boilers being injured. The witness stated that the hissing noise continued for about three minutes before the explosion occurred. Upon

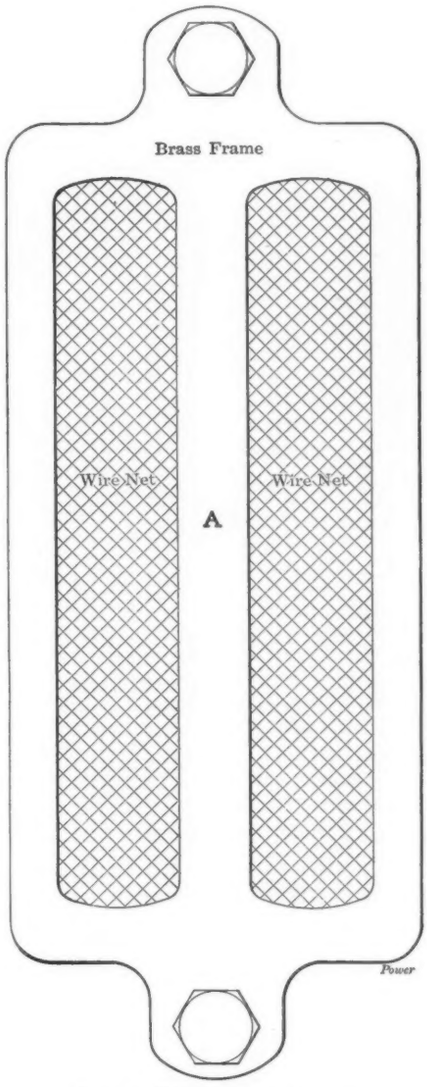


FIG. 1. WATER-GAGE GUARD

collapse the whole of the stoker mechanism attached to that furnace was wrecked and pieces, together with the doors and fire bars were hurled across the yard, demolishing part of a high wall, crossing the street, and through the third-story windows of a mill opposite. Live coals accompanied the wreckage and two spinning mules were ignited in the other mill. Several operatives were injured by the flying scrap and broken glass, practically the whole of the windows on the east side of this mill being broken by the force of the explosion. Thanks to the prompt action of the fire brigade the fire was soon subdued.

The low-water alarm had only been cleaned the day before the accident and was not heard to blow. The fireman, in his evidence, stated that the steam-pressure gage showed about 150 pounds and corroborated the statement of the dead man that water was within 3 inches of the top of the glass.

The stoker hoppers obstructed the view of the water-gage glasses from the sides of the boiler and the writer noticed guards in front of the glasses on the unharmed boilers, consisting of wire net mounted on brass frames. The shape of the frames was something like Fig. 1, and the center strip of brass *A* obscured the glass from the front of the boiler; the attendant would therefore have to be rather near the boiler to get a view of the glass between the stoker hopper and the brass guard.

In the last report of the insurance company it is stated that there was more deposit inside the boiler than was desirable. After the explosion some scale was found on the top of the flue close to the front head and this may have caused the small bag shown at *A*, Fig. 2. Scale anywhere near the rupture would have been swept away by the rush of steam and water at the explosion. Fig. 2 is a photograph of the collapsed flue taken from the front, and shows the rupture in the third flue ring. The edge of the break shows a perfectly clean edge and might indeed have been cut with shears. The thickness is drawn down at the break by the stretching of the metal at the edge, from ¾ inch, the apparent original thickness, to 9/16 inch. The split extends half way around the flue and no flaw or undue corrosion of the plate is discernible. The bagging took place directly over the fire, as will be seen by the location of the back supports for the grate. The plates show no signs of being burnt although the coroner, in summing up for the jury, said: "The approximate cause, as far as could be ascertained, was the overheating of the flue and perhaps there might have been another cause, the scarcity of water in the boiler."

After inspection, the writer is not inclined to support the latter theory in the face of the statements of the firemen,

the conditions of the low-water alarm and the unburned plates. It is, however, possible that the men were deceived in the water level.

At *B* in Fig. 2 is shown the lower right-hand water-gage connection. The corresponding left-hand connection is about a foot to the left. This connection was badly furred up and the small aperture available may have been blocked by a piece of scale at the time of the explosion. The fireman would stand at the left-hand side of the boiler to operate the blast and this is the gage he would be most likely to see at that time, although as has been pointed out the stoker hopper would be in the direct line of sight.

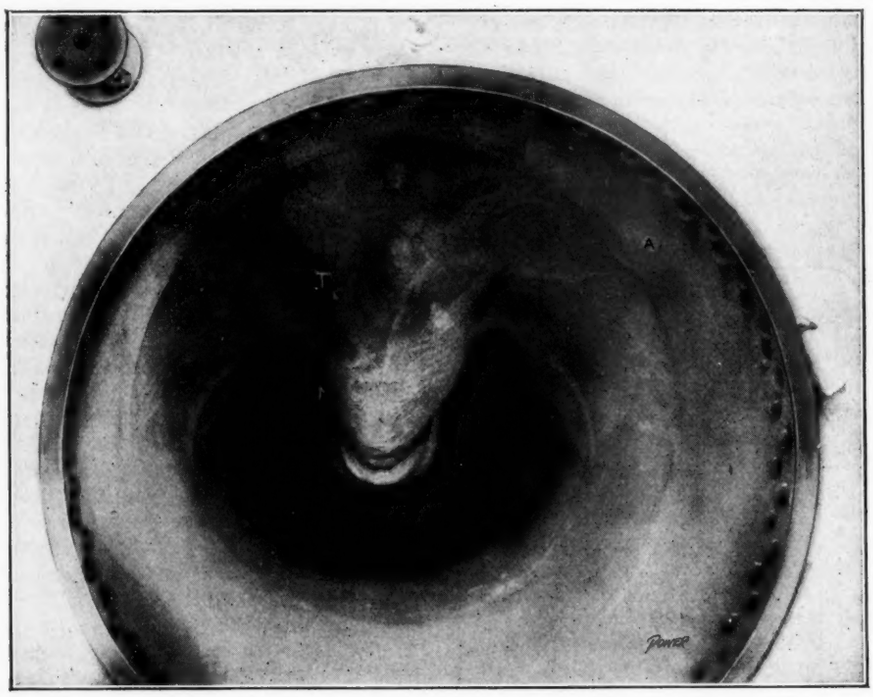


FIG. 2. VIEW OF COLLAPSED FLUE TAKEN FROM THE FRONT

If the low-water theory is unproved the writer would advance the following suggestions: The boiler is large, 8 feet 6 inches by 30 feet. The pressure authorized and sometimes worked at, 170 pounds, is high for a Lancashire boiler of this size, and necessitates a degree of stiffness in the end plates which might interfere with the proper expansion and contraction of the flues and cause undue stresses to be set up. The top of the flue expands more than the bottom because of the unequal action of the fire, and neglect of careful provision for this expansion institutes a severe racking action at the joints and tends to tear the plates. The boiler was off duty on Saturday at midday. Fires were relighted on Sunday evening. This means that for cleaning to have taken place in the interval, considerably quicker cooling of the boiler must have been resorted to than is at any time advisable. Taking the severest case, the hot water blown down under pressure and cold pumped in to cool the hot plates: The temperature corresponding to 170 pounds is about 370 degrees Fahrenheit. If the cold water is at 60 degrees the temperature range is 310 degrees. The coefficient of expansion of the metal may be taken as 0.00000672; the boiler is 30 feet long. The contraction, therefore, upon cooling equals

$30 \times 12 \times 310 \times 0.00000672 = 0.75$ inch. Assuming Young's modulus to be 30,000,000 for the material, the stress induced is

$$\frac{0.75 \times 30{,}000{,}000}{30 \times 12} = 62{,}500$$

pounds per square inch, or nearly 28 tons per square inch. While some of this is taken care of by the elasticity of the ring joints, it gives some idea of the enormous stresses which may be induced. This fact, combined with the rather unwieldy length of the fire grate, the possibility that the new stoker was overfeeding and the undoubted presence of scale on the flue may have been the original cause of the rupture. If once a slight bag be formed in a flue the pressure inside the boiler quickly forces it out until it assumes large dimensions. It would be difficult to observe the formation of any bags while the boiler was under fire, owing to the location of the stoking apparatus with relation to the furnace opening. One of the details hard to account for is the formation of the small bag *A*, unless it was due to scale alone, but even then it is above the coolest part of the fire. Feed water was introduced by a pump through an economizer. The shell of the boiler is apparently in the best of condition and all riveting is ample.

An unusual feature of the explosion, especially when the external damage is considered, is that the boiler has not stirred at all in its setting in any direction. Luckily the space front and back was ample, so that the force of the explosion was able to spend itself freely. If the wall at the back had been close up, as is too often the case, it would inevitably have been blown down and the boiler house would probably have collapsed.

Have POWER readers any suggestions to make as to the cause of this accident? In conclusion the writer wishes to gratefully acknowledge the facilities granted him by the mill management and the courtesy of the staff.

From a paper upon Steam Turbines in the navy, by M. E. Clergeau, a retired inspector of the French navy, published in the annual bulletin of the Société des Anciens Elèves de l'Ecole Professionnelle de Nantes, we have clipped the following cautionary suggestions in regard to the starting up of this type of prime mover. If the turbine is considered at rest, it will be seen that, contrary to the conditions which exist in the reciprocating engine where the steam-tight pistons intervene between the boiler and the condenser, the steam can pass freely from the boiler to the condenser through the zigzag passages of the wheel, and this condition will endure until the turbine gets into action and the pressures within it are established. This phenomenon lasts only a few seconds, but this time will be sufficient to bring on, if proper precautions are not taken, some disagreeable surprises: First, on the part of the boilers, the excessive draft of steam from which may produce a considerable lowering of the pressure and excessive priming, while the condenser may be gorged with steam and fail to maintain a vacuum.

The attention of the operator should be directed, therefore, particularly to the condenser. The circulation pumps should be started first of all at their maximum speed, as well as the air and small feed pumps. The fires should be burning briskly, in order to make up quickly the excessive preliminary draft of steam.

Aboard ship this condition is apt to be of particular importance in working the vessel into port. It is especially serious in the case of reversing, as, on account of the smaller number of elements in the reversing turbine, the steam has a still more direct path from the boiler to the condenser.

Water Power and Its Development*

Preliminary Considerations to Be Taken into Account in the Selection and Development of Suitable Sites and a Glance at Current Practice in Water Wheels and Their Installation

BY CHARLES EDWARD LUCKE†

Wherever rain falls, streams will form, the water of which represents the concentrated drainage of all the land sloping toward that particular valley at the bottom of which the stream flows. This stream flow consists of the rainfall over the whole watershed less the amount absorbed by the earth or evaporated from the surface, and every such stream is a potential source of power.

While all streams are possible sources of power, from which it might seem that water power was available all over the earth, yet, as a matter of fact, only those streams that are large enough or in which the fall of level is great enough, are really worth while to develop; and only in those districts where the rainfall is great enough and the earth not too flat or absorbent or the air too dry, may any streams of useful character at all be expected.

The power represented by any part of a stream is directly proportional to the product of the quantity of water flowing per second and the difference between the levels at the beginning and the end of that part under consideration, which difference in level is called the "hydraulic head." Any portion of a stream then, in which there is a difference of head, may be considered as available for power.

Whether it will be worth while to carry out a development will depend entirely upon the ultimate cost of the power produced, the market for that power and the cost of competing power generated from fuel. In every case the largest item in the cost of power per horsepower-hour or per horsepower-year is the fixed investment rate, representing the interest on the first cost of development. The definite yearly per cent. of this development cost divided by the power produced gives a yearly charge, which is the largest item in the cost of water power and may even exceed the fuel and labor cost in steam plants. It therefore appears that, assuming the stream to possess available power, it will be worth while to use that power only when the following conditions are satisfied: If the flow is large with perhaps moderate head or when the flow is small with large head, if the cost of diversion and concentration is small enough, if there is a sufficient power demand or selling market available and finally when the cost of steam or gas power is high in that particular locality.

It usually happens that the cost of development does not stop merely with diversion and concentration of the water at the wheels, because rainfall is irregular, and so, of course, is the resulting stream flow, which fact introduces a necessity for storage reservoirs. The size and cost of these storage reservoirs will depend upon the extent of the fluctuation of stream flow and upon the time intervals. A district in which there is a very heavy stream flow for a few days, followed by a month of no flow, would require a reservoir large enough to hold practically the whole flood water, so that it could be supplied uniformly to the wheels during the rest of the time.

No two streams representing power sources offer quite the same problem of power development, the first step in which is the reaching of an intelligent judgment as to whether or not it would be worth while to develop at all, which judgment must be based upon the most painstaking investigation of a multitude of conditions involving rainfall, stream flow, water storage and power demand, together with estimates of the probable cost of the power to be produced and the cost of competing steam or gas power.

Seldom does it prove worth while to attempt to develop all the power available at a given section of a stream, because the cost of the works is always large and if the installation is made large enough to use the maximum stream flow, then in time of drought or freeze-up, less power is available, and a part or perhaps all of the investment will be idle and earning nothing. But even worse than this, those industries dependent upon the power will be unable to proceed, thus rendering the capital invested in the industry, also useless. To protect dependent industries it will be safe to develop only so much power as is represented by the minimum stream flow. In cases where the industry demand for power exceeds this minimum supply then the remedy of storage reservoirs is available to a limited degree, after which steam- or gas-power auxiliaries must be installed, further complicating the problem. For when auxiliary fuel-burning power is used it naturally is idle in times of heavy stream flow, which fact increases the cost of the power delivered. Such difficulties as these all arise, of course, from fluctuating stream flow which is violent in some places and practically nonexistent in others. Streams are studied by engineers and government officers with regard to their flow and the results made public records.

The study of water-power conditions is very old indeed, but it is only for the past fifty years or so that these broad aspects of the question of economic development have received attention. The recent progress is due partly to the fact that in the early days there were no wheels capable of using high heads, and partly to the lack of a market for power, but very largely because many of the most effective water-power sites were located far from the towns and with no transportation facilities between. With the increasing demands for power, attention was directed to other water powers than those early developed and a better understanding of the principles of design enabled engineers to build suitable high-head water wheels. The greatest factor of all, however, in this new and vigorous utilization of water power came from the progress in electrical engineering, which showed how the power of rotating shafts could be converted into electrical energy. Such electrical energy, without prohibitive losses or excessive costs, could be made to flow on wires over distances exceeding 100 miles and at the end of a transmission line the electrical energy could be converted back into the form of rotating shaft power by motors or converted directly into light by electric lamps. By this electrical transmission of power, progress in water-power generation and use has been stimulated to an extent that would have otherwise been impossible, for it is far cheaper to run wires over the mountains to towns already located and in communication with the rest of the world by river or railroads, than it is to run railroads through mountainous countries to permit of the establishment of a factory at the water-power site.

As a result of this latter period of economic development the practice has more or less crystallized, and in reviewing the situation as it exists today, it is convenient to divide those cases that have proved to be worth while into three typical classes with regard to the head, assuming, of course, that this head is available in not too great a horizontal distance. The first class includes low heads of 20 or 30 feet, the second class includes medium heads between 100 and 200 feet, and the third class, heads up to or exceeding 1000 feet. In no case must the horizontal distance to secure this head be too great and in practice it has been found that the abrupt head, such as oc-

*Seventh Hewitt lecture delivered in Cooper Union hall, March 21, 1910.

†Professor of mechanical engineering in Columbia University.

curs at falls of considerable hight, is almost ideal, while a horizontal distance of two or three miles is not out of the question. Perhaps the best known large-flow, abrupt-drop, medium-head site is that of Niagara Falls. The fall varies in hight from 158 to 167 feet and the width of the crest is about ½ mile, over which the volume of the flow is not exactly uniform. It is estimated that about 7 million horsepower is here available, of which only a small portion is developed or under consideration. Another example of a still greater falls of much the same character as Niagara is the great cataract of the Zambesi river in Africa, still undeveloped because of lack of demand for power in that locality, although many projects have been formulated, some of which will ultimately be carried out. Here the hight is a little over 400 feet and it is estimated that in time of flood over 35 million horsepower may be secured, but by reason of the tropical and irregular character of the rainfall only a small fraction of this could be depended upon for steady generation.

Extremely high heads are available only in the mountains and obtainable only by more or less lengthy horizontal distances. Development in these cases will, therefore, cover some miles of mountainous country, often involving the boring of rock tunnels and the construction of canals carrying water around the sides of hills. An instance of this kind is the plant of the San Joachim Electric Company, of Fresno, Cal. To develop a head of 1400 feet a pipe line 4000 feet in length is fed by a canal seven miles long, a condition which would never have been justified unless so great a head could be available.

The last of the three classes of streams is the low-head class, where either at a low natural water fall or at the head of a section of the river, a dam is placed, fixing a high level for the water, the low point being located at some convenient spot down stream. An instance of this is the Susquehanna river, recently developed at McCall's Ferry.

In the early days of American development only low heads were attempted, and on the sites chosen, cities grew. These low heads were selected because of the prevailing form of water wheel, which was the familiar old-mill overshot or breast type, completely exposed, and of a diameter approximating the hight of the falls. These wheels were located first at either the side of a natural fall or at one end of a dam built above a fall or rapids. When more power was wanted another mill with its wheel was located at the other end of the fall or dam and when more than this supply was wanted there began canal construction. According to Col. Samuel Webber, the first water wheel to be erected in this country was built in 1790 at Pawtucket Falls for cotton manufacture and was soon followed by others at Paterson, N. J.

Perhaps the first tendency in the changing of conditions was the result of steam competition which appeared in 1830, at which time the first steam-engine mill was erected at Providence, R. I., and it is an open question whether this competition did or did not tend to stimulate the study of water power; but whether it did or not there was applied fourteen years later (1844) at the Appleton mills, in Lowell, the first hydraulic turbine, which type makes use of the spouting capacity of water under pressure as it issues from the pipe. Here the water jet acts upon curved vanes, operating them by giving up its velocity; an action which distinguishes them from the old and most common overshot wheels, where the water turns the wheel by its weight alone. Modern water wheels designed with nozzles and curved vanes act by both reaction and impulse and sometimes by both in the same wheels, and are no more than refinements of these simple affairs which started a movement completely revolutionizing water-power development. The largest turbines in existence are four built by I. P. Morris Company, for the Great Western Power Company. These are operated on a vertical shaft and develop 18,000 horsepower each, at a head of 525 feet, and with nearly 90 per cent. efficiency.

The many types make it possible to either lead the water to the turbines through pipes or submerge the turbine in the head waters above the dam; and they may be run with vertical or horizontal shafts. The direction of flow may be up or down, in or out, in practically any convenient way, making their application by reason of this flexibility a comparatively easy matter. In all cases the water acts over the whole circumference of the wheel and whether it enters through guide vanes to ultimately discharge through nozzles, as in reaction wheels, or enters nozzles directly, to strike vanes later, as in impulse wheels, the nozzles themselves form a continuous ring and are made by inserting partitions of suitable form between the plates.

To regulate the quantity of water passing through these nozzles, gates are always provided, and a water wheel that is not provided with gates is practically useless for electrical operations. When gates are closed to reduce the quantity of water passing through a wheel, the whole quantity of water in the pipe must be reduced in speed. This water is practically incompressible and the slowing up of this long column with its great inertia, without danger, is just as difficult a proposition as stopping a high-speed train. To avoid this difficulty relief valves are sometimes used, similar to safety valves on boilers, but they are not considered as reliable as another device, which, however, cannot be used on very high heads. This other device is a standpipe projecting up into the air near the power house. Under ordinary flow conditions the water would not run out at the top, but should the flow of water be suddenly arrested the water will spout out at the top.

The horsepower of a turbine is some fraction of the product of the quantity of water flowing and its head or pressure. This fraction is the efficiency or the ratio of the power that the turbine can develop to the energy of the water passing through, and in good wheels this is about 80 or 90 per cent. To control the speed of these wheels and at the same time to avoid setting up heavy inertia pressures in the pipe line by varying the quantity of water, it has become customary to install a governor which will deflect the nozzle away from the buckets when the speed gets too high. This allows the regulation of speed desired without varying the quantity of water, and incidently injuring the pipe.

Wheels being available for any head, however great or small, and a locality likewise available for a flow of water with a suitable head that can be developed in a reasonable distance, the problem becomes how best to adapt the wheel to the locality, or how best to modify the locality to receive the wheel. The modification in every case involves concentration of the flow, the providing of storage to take care of fluctuations, and the providing of safeguards against destruction. Unless there is a direct fall there must be a dam, the object of which is to fix the high level and to store water. In connection with the dam there will be at some point in the stream a spillway over which waste water may flow without injuring the plant. At Niagara, the falls themselves constitute a natural spillway, whereas at McCall's Ferry, there was required a half-mile spillway dam. Dams are made in all sorts of ways, from earth and rock reinforced by timber to masonry or concrete, running down to bed rock, and may, therefore, be cheap or expensive.

From above the dam or natural water basin, the water is led to the wheels in a great variety of ways. The wheels may even be within the dam itself, the casing being removed from the supply side and the inlet orifices exposed to the standing water; or the water may be led through wooden flumes, masonry canals, rock tunnels or vertical shafts bored in the rock or through iron or wooden pipes. Somewhere in the open conduit there is usually provided a sort of basin called a forebay, to which the head gates are attached so that the water may be shut off during repairs. There will also be placed at this point rubbish or ice racks to intercept floating ice or logs. Whenever there is an upper level canal it is called a headrace. Below the wheels a canal is needed sometimes to reach the lower level waters and is called a tailrace.

English Practice in Condensing Equipment

The Contraflo Condenser, Parsons Vacuum Augmenter and Recent Developments in the Jet Condenser of Single and Multijet Ejector Types

BY JAMES A. SEAGER

One of the most important developments in recent years as regards condensing practice is the introduction by the Contraflo Condenser Syndicate of their special design of condenser which is illustrated and described in the February 16, 1909, and the June 1, 1909, numbers of POWER AND THE ENGINEER. The importance which the Contraflo condenser has attained in British electric-power house practice may be gaged from Table 1.

Among the very large number of power houses which are equipped with Contraflo condensers may be mentioned the Grove road power house of the Central Electric Company of London. Two rectangular condensers made by Richardsons, Westgarth & Co., Ltd., work in connection with exhaust-steam turbines, the quantity of steam dealt with by each condenser being 40,500 pounds per hour. The cooling surface of each condenser, which is shown in Fig. 1, is 3350 square feet, and a vacuum of 27½ inches is maintained with the barometer at 30 inches, the temperature of the cooling water being 70 degrees Fahrenheit. Another instance is that of two circular condensers supplied by the same firm to the Manchester Corporation and working in connection with 5000-kilowatt turbine sets. The total cooling surface of each condenser is 10,000 square feet, and the steam dealt with is 100,000 pounds per hour. The vacuum maintained with a 30-inch barometer is 27 inches, the temperature of the cooling water being 85 degrees Fahrenheit. Fig. 2 shows one of these condensers.

Another important installation of this type is at the Carville power house of the Newcastle-on-Tyne Electric Supply Company, Ltd., where five Richardsons-Westgarth condensers are installed in connection with Parsons turbines. These, according to figures kindly supplied by Merz & McLellan, the consulting engineers to this system, are each of 10,500 square feet cooling surface, and deal with about 64,000 pounds of steam per hour, the kilowatt capacity of each generator being about 4000 kilowatts. The square feet of cooling surface is 10,000 and the air pumps are of the three-throw Edwards type. The performance of these condensers may be judged by a test result obtained in 1907. The test measurements lasted for one hour, the mean calibrated kilowatts being 5164.07. The vacuum obtained at the turbo exhaust, with the barometer at 30 inches, was 29.039 inches, the pounds of water condensed per hour being 68,180.

THE VACUUM AUGMENTER

Very recently, owing to the increased use of turbines in power houses, these requiring a very high vacuum, a considerable advance has been made by the introduction of the vacuum augmenter for surface condensers. This, which is the invention of C. A. Parsons, is not only manufactured by C. A. Parsons & Co., of Newcastle-on-Tyne, but licenses have been taken out by important engine builders, such as Willans & Robinson and others. Fig. 3 shows the general arrangement of this apparatus as applied to a 2000-kilowatt condensing plant at the borough of St. Marylebone electricity works, *A* being the main exhaust pipe, *B* the main condenser, *C* the circulating pump, *D* the augmenter condenser, *E* the vacuum augmenter itself, *F* the seal pipe in the pump suction, *G* the air pump, *H* the circulating-water discharge, *J* the pump inlet, *K* the sluice valve and *L* the turbo-generator.

Fig. 4 is a diagrammatic arrangement of the vacuum augmenter. The apparatus includes a steam jet placed in a contraction of a pipe between the condenser and air pump, this drawing the air and vapor from the condenser and reducing the density of the vapor in it to about one-third its normal value. In the neck of the pipe the mixture of air and vapor is compressed to about 50 per cent. of its bulk, and a small condenser, of cooling surface of about 2 or 3 per cent. of the main condenser, is interposed between the neck and the air pump. The air is thus cooled and the vapor partly condensed before it reaches the air pump. The condensed water falls by gravity directly from the main condenser to the air-pump suction by a water-sealed connecting pipe. In recent practice the steam

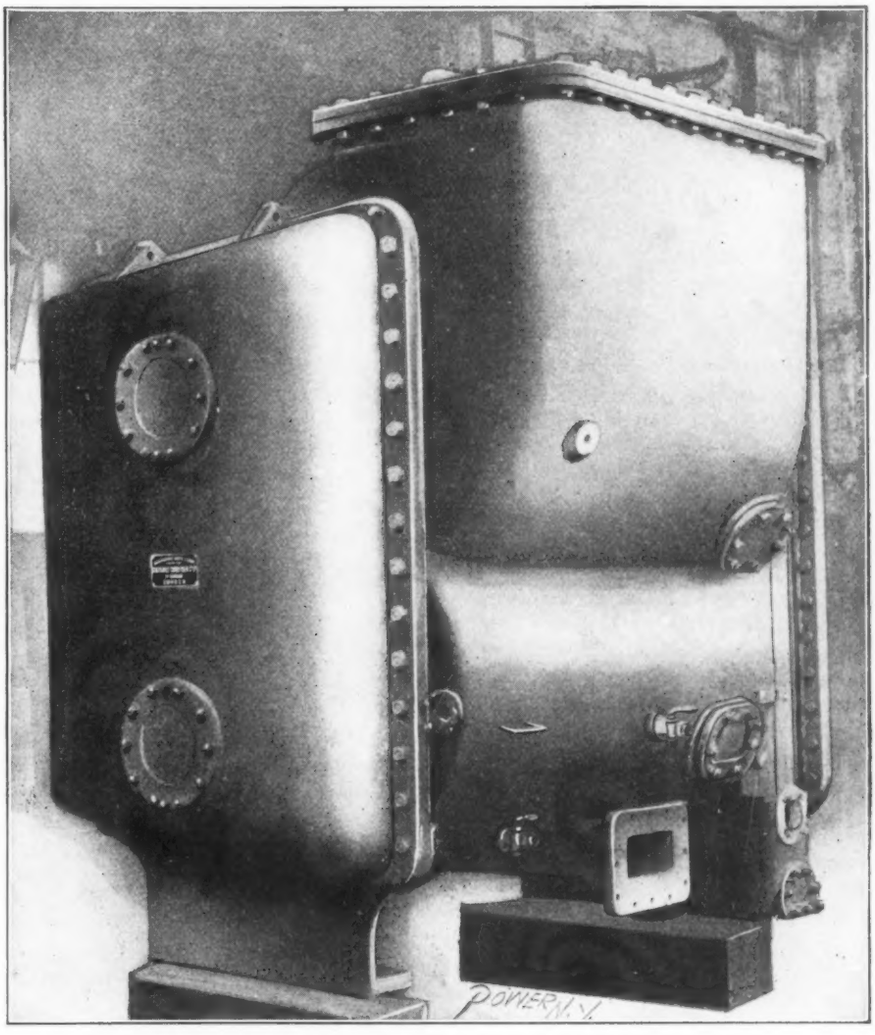


FIG. 1. CONTRAFLO CONDENSER AT GROVE ROAD POWER HOUSE

TABLE 1. CONTRAFLO CONDENSERS IN BRITISH POWER PLANTS.

Power House.	Type of Main Engines.	No. of Condensers Supplied.	Cooling Surface in Sq.Ft. in Each Condenser.	Amount of Steam in Lb. Per Hour.	In. of Vac. with Bar. at 30"	Inlet Temperature of Circulating Water in Degrees F.
Manchester Corporation	Parsons turbines.	2	10,000	100,000	27	85
Newcastle-on-Tyne Electric Supply Co	Turbine.	1	13,000	81,500	29.1	50
Glasgow Corporation	Parsons turbines.	2	8,000	73,600	28	70
Sheffield Corporation	Parsons turbine.	1	8,000	73,332	28.5	60
Newcastle-on-Tyne Electric Supply Co	Parsons turbine.	5	10,500	65,000	28.8	55
City & South London Railway Co	Reciprocating engines.	1	3,800	50,000	26	85
Edinburgh Corporation	Combined reciprocating engine and turbine.	4	3,750	45,650	27.5	60
Central Electric Supply Co., Ltd., London	Combined reciprocating engines and Parsons turbine.	2	3,350	40,500	27.5	70
Spennymoor Station of Northern Counties Electric Supply Co., Ltd	Parsons turbine.	2	4,400	38,000	27	83
Bury Corporation	Turbine.	2	4,300	33,000	28	65
Dundee Corporation	Parsons turbine.	2	3,400	32,600	28.5	60
Hammersmith Corporation, London	Parsons turbine.	1	3,750	27,000	28	65
Dublin Corporation	Parsons turbine.	1	2,150	25,000	28.5	55
Stepney Corporation	Parsons turbine.	2	2,200	20,000	28.5	60
Lisbon Tramways Co	Parsons turbine.	1	3,050	20,000	28	
Dewsbury Corporation Electricity Works	Reciprocating engine.	1	1,700	20,000	26	85
Southampton Corporation	Parsons turbines.	1	2,100	18,375	28	70

consumption of the jet is from 1 to 1½ per cent. of the normal amount of steam passing to the main condenser, and the net reduction of steam consumption in the turbine set (for which purpose the vacuum-augmenter sets are chiefly used) is about 8 per cent. at full load, and the speed of the air pump as before. The importance of this development may be measured by the following considerations put forward by Messrs. Parsons and Stoney in their paper before the Institute of Civil Engineers on the steam turbine:

Under usual conditions a vacuum of 26 to 27 inches can be obtained in a good condenser having about 1 square foot of surface per indicated horsepower and circulating 30 times the feed at a temperature of about 70 degrees Fahrenheit. In order to obtain a vacuum of 27½ to 28 inches and thus to effect a saving of 5 per cent. in coal consumption, the condenser surface has to be increased by about 50 per cent.; the velocity of the water through the tubes must rise to 4 to 7 feet per second, giving a turbulent flow and better heat absorption, this being obtained by increasing the volume of the circulating water and modifying the condenser design. The flow of steam among the tubes should be easy and the condensed water must be quite cool before reaching the air pump, even if the bottom two or three rows of tubes are drowned, and probably a larger air pump is necessary. The curve, Fig. 5, shows clearly the effect of increased vacuum on the output of a Parsons high-pressure steam turbine using 30,000 pounds of steam per hour at 175 pounds stop-valve pressure. The full curve shows the net output at various degrees of vacuum, and the dotted curve the gross output. The difference between the two shows the total power absorbed by the air-pump motor, the circulating-pump motor and the steam in the augmenter jet. An allowance of 15 feet head of water above tube friction in the condenser is also made.

An example of vacuum augmenter is to be found at the borough of St. Marylebone electricity works, London, where two Parsons surface condensers having vacuum augmenters take the exhaust steam from 2000-kilowatt units. The condenser is placed immediately under the exhaust chamber of the turbine. The air pumps are of the three-throw type, with a hotwell force pump attached, driven by a 17-horsepower motor. The air pumps and motors are placed in basement spaces between alternate pairs of generating sets. As the result of tests the following figures are obtained: The cooling water at the inlet of the condenser was 76.5 degrees Fahrenheit, rising to 89.5 degrees Fahrenheit at the outlet. With the barometer at 30 inches a vacuum of 28.33 inches was obtained when the pounds of water at the air-pump discharge were 35,155.

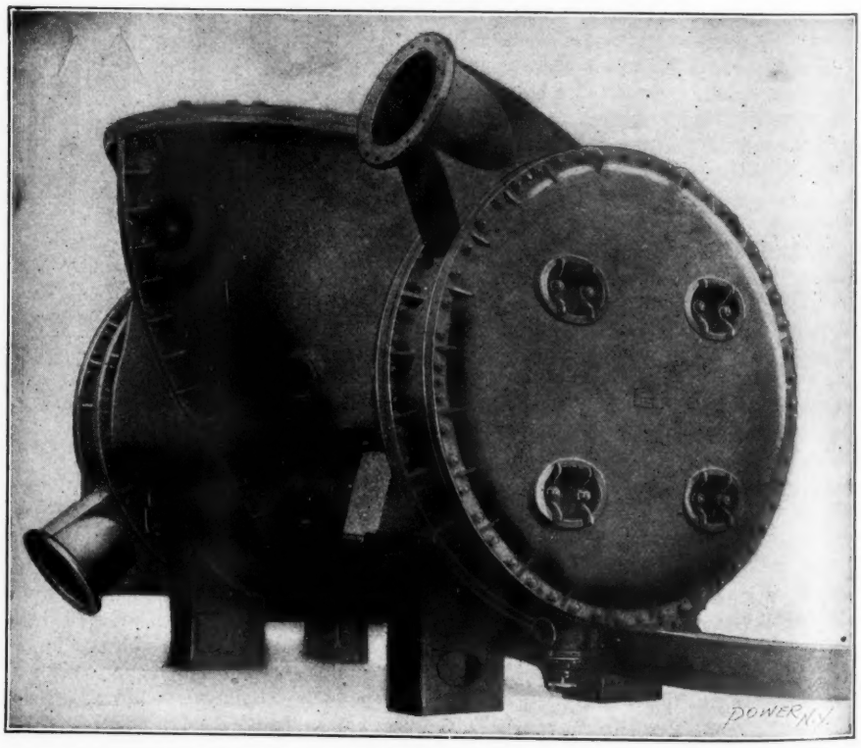


FIG. 2. CONTRAFLO CONDENSER SUPPLIED TO MANCHESTER CORPORATION

The temperature of the vacuum maintained was 95.3 degrees Fahrenheit and the hotwell temperature 89.8 degrees Fahrenheit. As the cooling surface in square feet of the condenser was 3200, the pounds of water per hour condensed per square foot of surface area reached the high figure of 11 pounds.

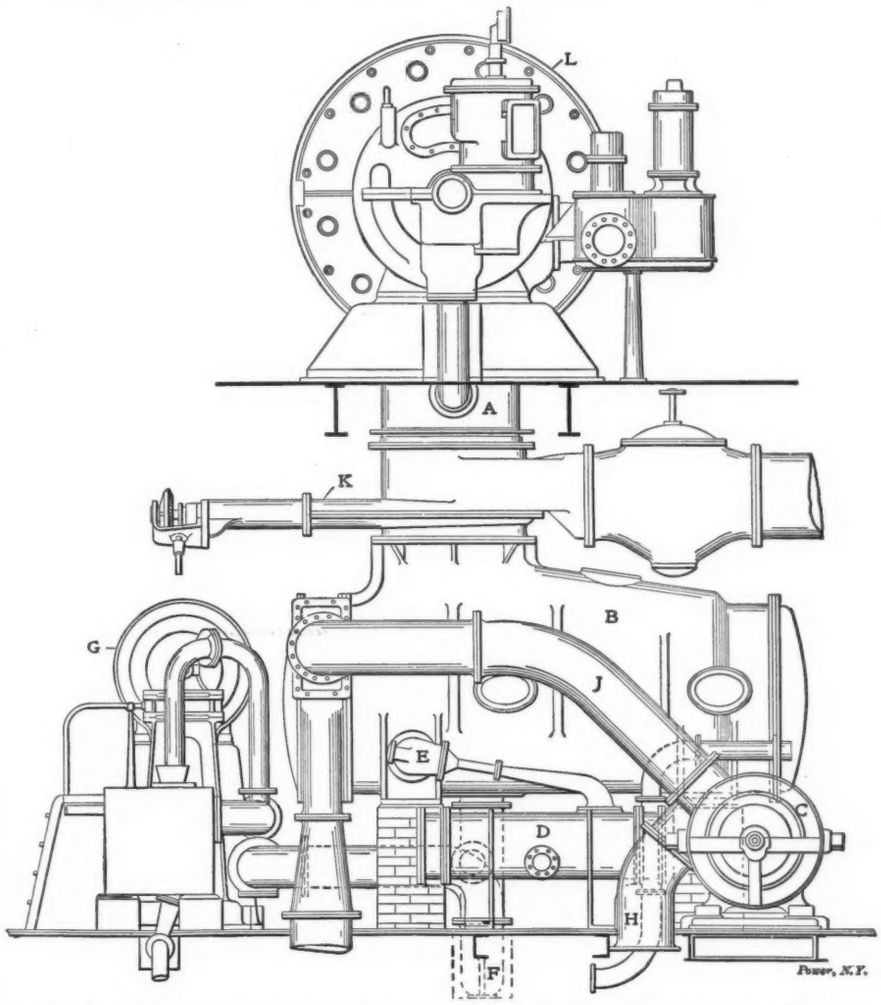


FIG. 3. VACUUM AUGMENTER APPLIED TO 2000-KILOWATT CONDENSING PLANT

THE JET CONDENSER

With the advent of the turbo-generator and the absence of oil in the condensed water, the jet condenser in one of its improved modern forms has entered into favor. The countercurrent type, in which the direction of the steam in the inclosed vessel is generally in an opposite direction to that of the cooling water, and in which the contained air is drawn from the coldest place in the condenser, close to the entrance of the injection water, is now becoming popular in English practice. These plants are either of the low-level type, in which the air pump does work against pressure in drawing away the condensate and the air contained in the steam; or of the barometric type in which the condenser itself is elevated above the pump, the water being delivered into the condenser and not sucked from it, the column of water in the discharge pipe, on the other hand, constituting a head by which the water is drained away against vacuum. Suppose the injection branch is raised 40 feet above the pump and the vacuum obtained is 26 inches, the actual head is

$$40 - 29.5 = 10.5 \textit{ feet},$$

a gain of 52 feet head over the low-level type. With a vacuum of 28 inches the actual head is reduced to

$$40 - 31.7 = 8.3 \textit{ feet}.$$

A good example of the low-level type is furnished by the "Zylba" patent jet-condensing plant made by the Midland Engineering Company, and shown in Fig. 6. It will be seen that the general course of entrance steam is in opposition to that of the water. A partition (shown dotted) up one side of the chamber allows the air to be drawn from the top, or coolest part, and separate outlets are provided for the air and water discharge. Fig. 7 shows the arrangement of a barometric jet-condensing system in conjunction with a cooling tower installed by the same firm, while Fig. 8 gives a general view of a condenser of similar general type built by the Worthington Pump Company, Ltd. It will be noted that in this the air is drawn from the top of the condenser past a water pocket which catches any condensed moisture that may be drawn over with the air. At the present time the Midland Engineering Company is constructing a barometric jet condenser for a power house, whose exhaust pipes are 4 feet in diameter. By means of this type, countercurrent condensers can be used with only one pump, thus saving a considerable amount of circulating water and power for driving the pumps without increasing the first cost of the plant above that of an ordinary parallel-current plant.

Where, owing to the use of reciprocating engines or other causes, the presence of oil is anticipated in the exhaust steam, an oil separator can be combined with the barometric condenser. A very fine example of barometric countercurrent jet-condenser plant is that recently installed by the Worthington Pump Company, Ltd., at the Neasden power station of the Metropolitan Railway Company,

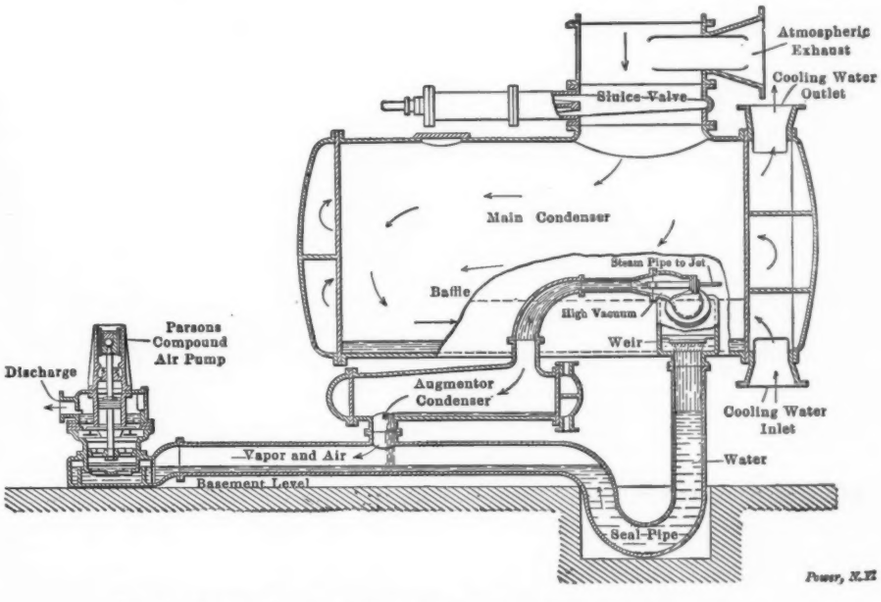


FIG. 4. USUAL ARRANGEMENT OF PARSONS VACUUM AUGMENTER

London. This is shown in Fig. 9, and has a capacity of 80,000 pounds of exhaust steam per hour at normal and 120,000 pounds at maximum load.

Multijet Condensers

A very interesting type of jet condenser which is coming into a consider-

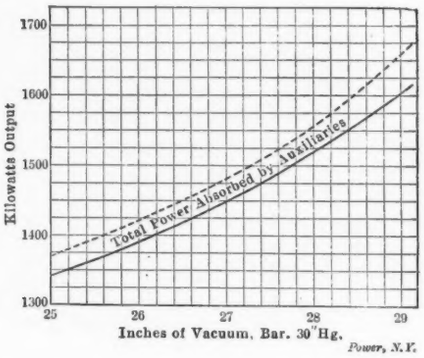


Fig. 5. Increased Output from High Vacuum

able degree of favor in English power houses is the multijet ejector condenser made by Kőrting Brothers, Ltd., of Westminster. For many years they have made injector condensers, giving excellent results in comparison with other types of condensing plant, for vacua up to 26 inches, but this apparatus has not compared favorably with surface condensers for higher vacua requirements. They have therefore been induced to make a considerable number of experiments with various forms of ejector condensers, and have succeeded in designing an apparatus which gives better results than can be obtained with condensers of the older type. The construction of the new condensers is clearly shown by the section in Fig. 10. With the older type of ejector condenser a single jet is used, and this passes through a long cylindrical tube provided with a series of rings or holes for the passage of the exhaust steam from the condenser chamber to the jet. The holes are drilled obliquely, directing the steam at a suitable angle so that it impinges on the condensing jet in the direction of flow. The steam, which strikes the condensing jet at high velocity, is condensed, and the particles of water into which it is converted, having the kinetic energy due to the steam velocity, cut into the jet and contribute to the momentum needed to discharge this and the entrained air and noncondensable gases against the resistance of the atmosphere. The improved condenser works on the same principle, but has, instead of one central condensing jet, a number of converging jets meeting and forming a single jet in the lower part of the tube. This tube is cast in one piece and consists of a series of concentric nozzles of gradually diminishing bore. The stream flows through the annular passages between the nozzles which guide it, so that it impinges at a suitable angle on the condensing jets.

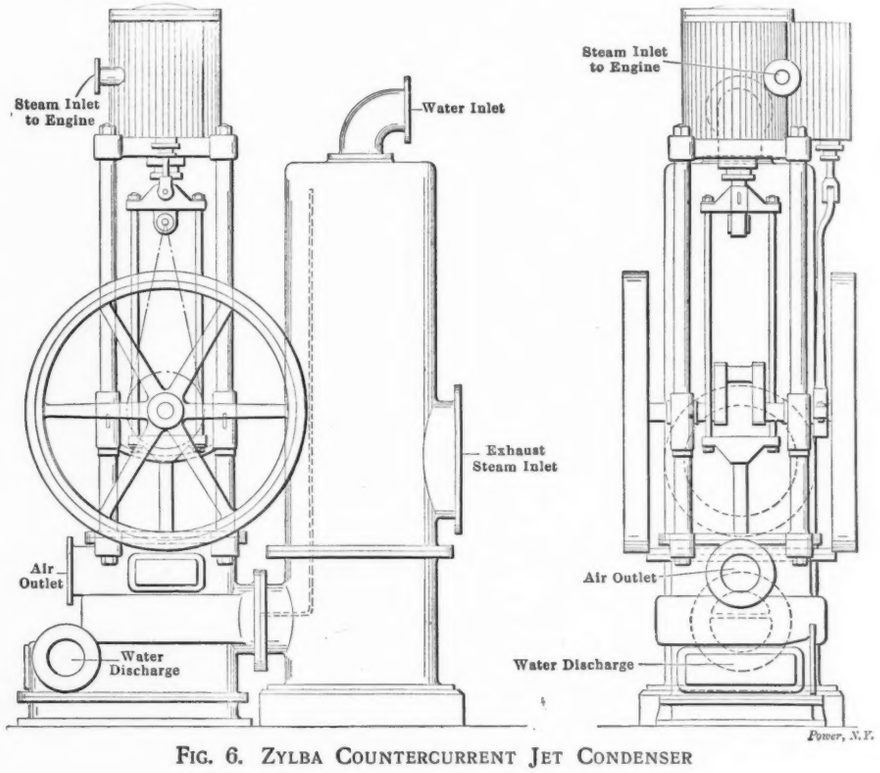


Fig. 6. Zylba Countercurrent Jet Condenser

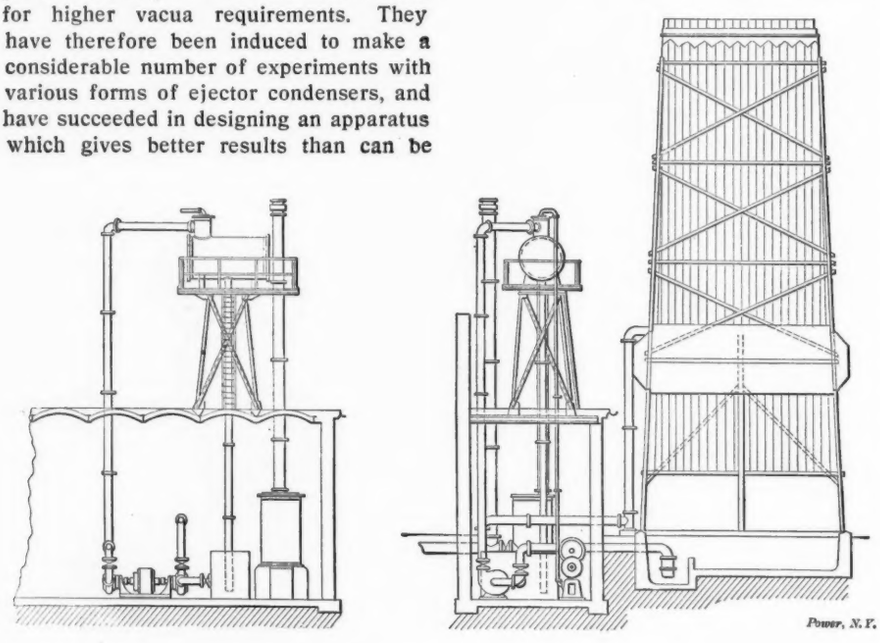


Fig. 7. Barometric Jet Condenser and Cooling Tower

The multijet condensers are considerably shorter than the single-jet apparatus of equal capacity, but in spite of this the area of contact between the steam and the water is greater. A further advantage is gained by the form of the condensing tube, which in vertical section is an inverted cone. In the upper part of the tube the steam is in contact with the coldest water and condensation is keenest, so that a greater weight of steam is condensed per unit of area of contact than is the case in the lower part of the tube, where the water is hotter. It will be seen that with the conical tube the sectional area of the steam passages increases from the bottom upward, and this secures that the sectional area of the ports is proportional to the volume of steam they have to pass. Owing to this, the steam velocity is nearly constant from the top to the bottom of the tube, and the exhaust chamber can be reduced to a minimum. It is clear that there must be a higher absolute pressure, or, in other words, lower vacua in the exhaust chamber than in the interior of the tube, or otherwise there would be no flow of steam through the ports to the condensing water jets. A drop of vacuum equal to ½-inch mercury column suffices with a multijet ejector

maintaining 28 inches of vacuum, and the actual working vacuum obtained is, therefore, only ½ inch lower than the highest theoretical vacuum, as determined by the discharge temperature and corresponding vaporizing point of the condensing water.

With the new condensers the ratios of water to weight of steam condensed appears to be, for equal vacua, practically the same as is required for surface condensers, and there is no ground for the objection often raised that ejector condensers require a disproportionate amount of condensing water. The multijet condensers have the same advantage as the older single-jet apparatus, that the vacuum is obtained and maintained without the use of air pumps.

To insure satisfactory working under all conditions of variation of load on the turbines or engines to which they are attached, it is only necessary to supply the water to the condensers at a pressure at the level of the water inlet flanges, equal to 20-feet water column, or say, 9 pounds per square inch. When, as is usually the case, there is no gravitation supply of water available, it is necessary to use a circulating pump. The power required for working the condensers is low and compares favorably with that needed for other forms of jet condenser with air pumps, or for jet condensers or surface condensers requiring water-circulating and air pumps.

Taking as an example a 1000-kilowatt reciprocating set, using at full rated output 20,000 pounds of steam per hour, a multijet condenser, using 55,000 gallons of water per hour, would maintain 26 inches of vacuum when dealing with this weight of steam with water supplied at a temperature of 60 degrees Fahrenheit and the barometer at 30 inches. The condenser would be 6 feet long and the water would have to be delivered at a pressure equal to 20 feet head at the level of the inlet flange. The lift for the circulating pump would be therefore 6 feet plus 22 feet plus allowance for friction losses and difference of level between the pump intake and the condenser outlet flange. Assuming an allowance of 6 feet would suffice for these last items, the pump duty would be 920 gallons per minute through a total lift of 32 feet, representing nine water horsepower. The combined efficiency of a motor-driven centrifugal pump should not be less than 60 per cent., and the power required would be 11 kilowatts of 1.1 per cent. of the full load output of the set. With a turbine of the same size requiring a vacuum at full-rated output of 28 inches, a condenser using 88,000 gallons of water per hour would be needed, and the power required would be

$$1.1 \times 88/55 = 1.8 \textit{ per cent.}$$

With an exhaust-steam turbine of the same capacity, using, say, twice the weight of steam per kilowatt output, the power required for working the condenser with 28-inch vacuum would be 3.6 per cent. of the full-load output. With circulating water at a temperature of 70 degrees Fahrenheit, the power required, other conditions being as above, would represent about 2 per cent., 5 per cent. and 10 per cent. of the full-load outputs, and at 75 degrees Fahrenheit about 3, 7½ and 15 per cent. If recooled water has to be used for condensing, it is in the last degree important that efficient cooling arrangements be adopted, as with water at a temperature above 75 degrees Fahrenheit the quantity of circulating water and the power required for working the condensing plant becomes disproportionately high. The multijet ejector condensers have no moving parts and wear and tear is nil even if dirty river or canal water, which would give trouble with surface or ordinary jet condensers, be used.

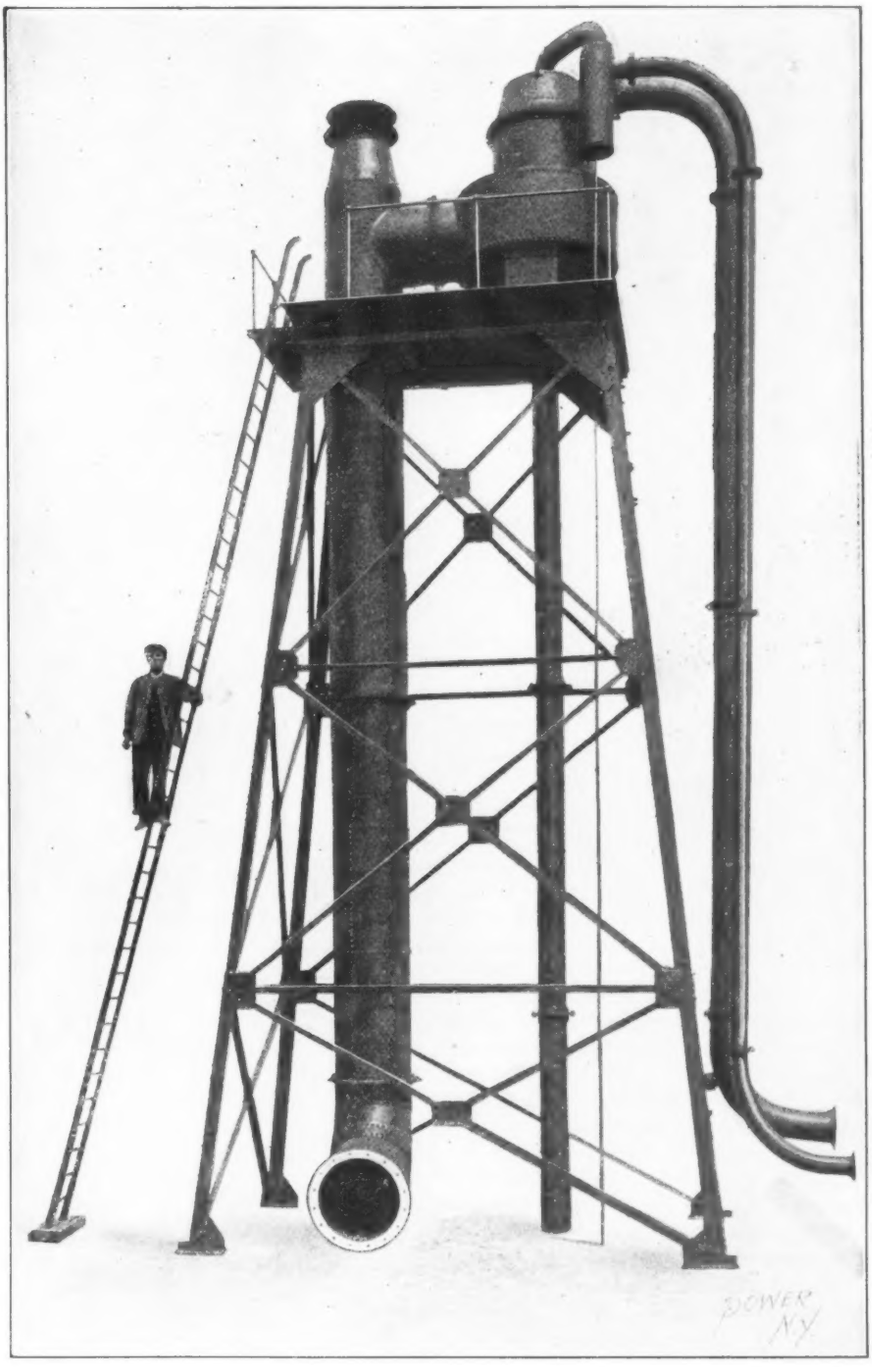


FIG. 8. WORTHINGTON BAROMETRIC CONDENSER

Fig. 12 shows the Kőrting multijet condenser which was installed at the Franco-British exhibition in London two years ago and was attached to a Parsons 1800-kilowatt turbo-tandem generator. For the complete arrangement, including their motor-driven centrifugal pumps, this firm was awarded the gold medal. This con-

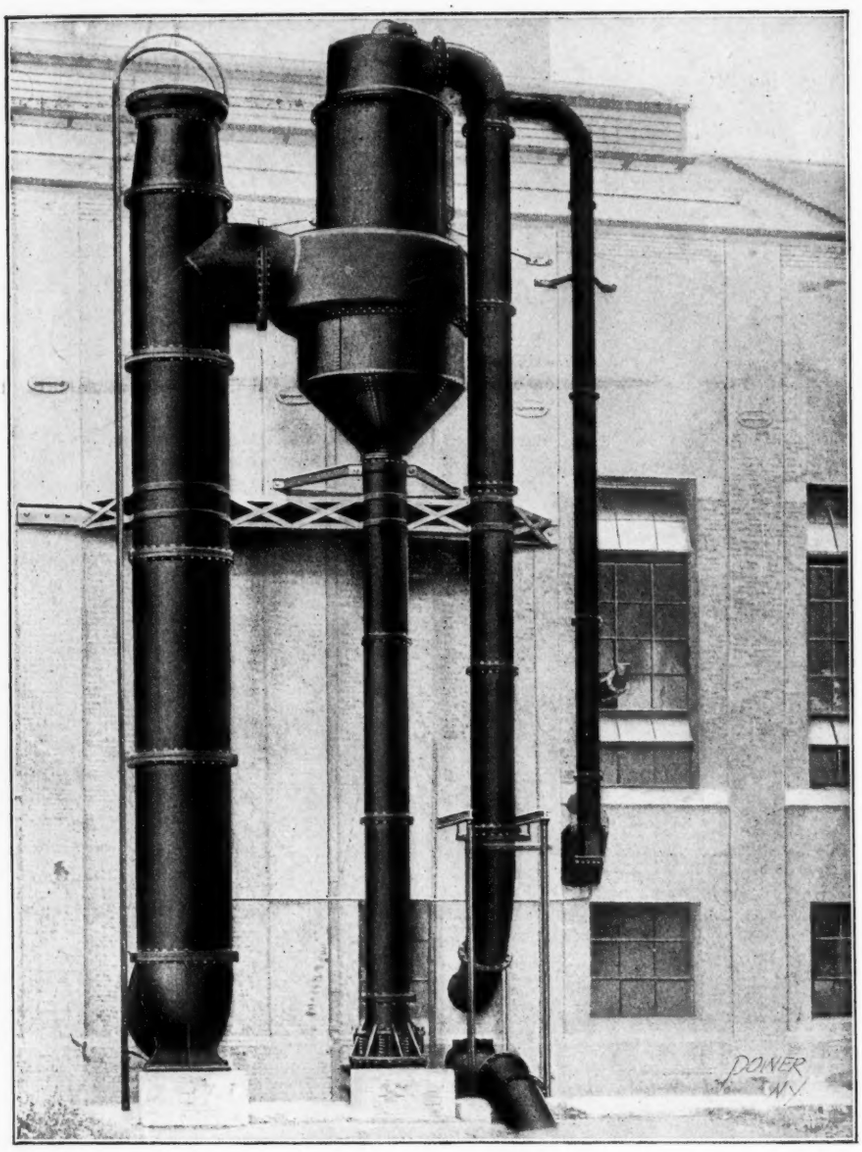


FIG. 9. LARGE BAROMETRIC CONDENSER OF THE METROPOLITAN RAILWAY COMPANY, LONDON

denser is quite new and is only now going into some of the large power houses, so that definite figures as to performance, beyond the fact that they are giving great satisfaction, are not available at present as regards turbine practice. The test figures in Table 2 were obtained on a 500-kilowatt generating set driven by a reciprocating engine at Hanley in Staffordshire during December, 1908, the vacua readings being taken by a mercury column at the condenser exhaust chamber.

It can reasonably be expected that with a high- or low-pressure steam turbine better results would be obtained. Fig. 13 shows the arrangement of pipework and plant adopted in attaching a multijet ejector condenser to a steam turbine.

There are in Great Britain a large number of power houses having reciprocating engines of medium size to which Kőrting ejectors of the older type described above are attached, as indicated at *A* in Fig. 11. From a list of 50 public electric-light and power-supply stations in Great Britain using such condensers we may select as example, Hull, where five ejectors dealing with 55,000 gallons of

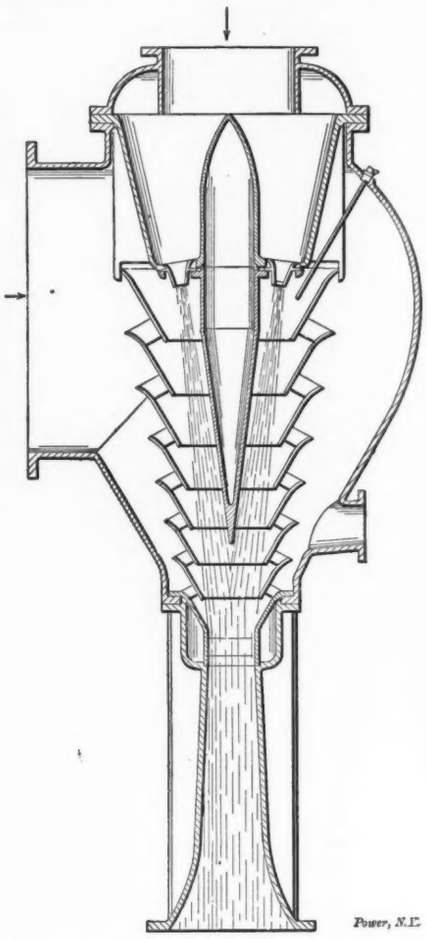
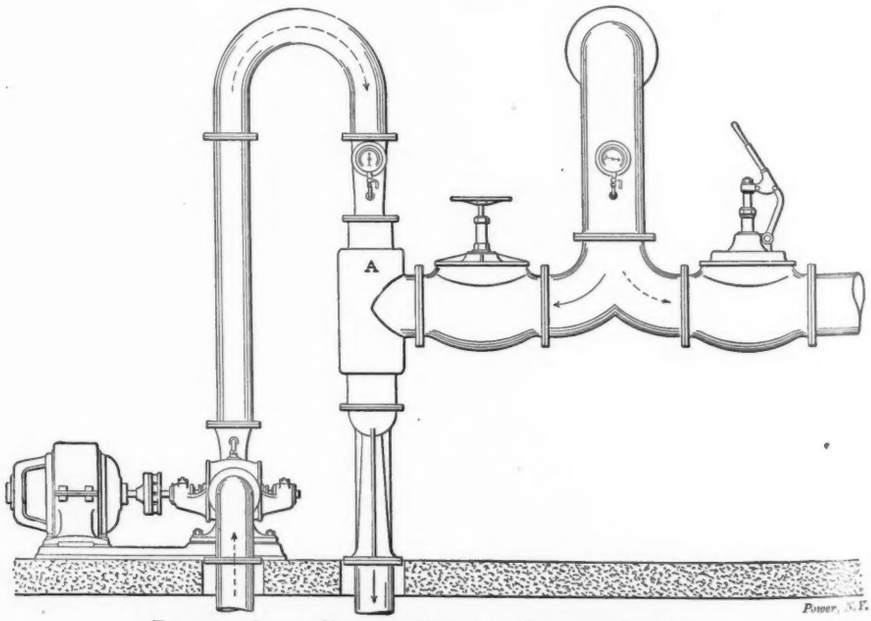


FIG. 10. KOERTING MULTIJET CONDENSER

circulating water per hour each, four with 38,500 gallons, two with 15,400 gallons, two with 14,300 gallons, one with 13,200 gallons and two with 3960 gallons, are in-



.FIG. 11. OLDER TYPE OF KOERTING EJECTOR CONDENSER

stalled for a station capacity of 5000 horsepower; Liverpool, where 13 ejectors each dealing with 55,000 gallons of circulating water are installed for 12,000 horsepower; Manchester, where 10 ejectors dealing with a total of 563,000 gallons of condensed steam for 9000 horsepower of electric generators; Wigan, where 4000 horsepower are dealt with by nine ejectors; Paisley, where eight ejectors cope with 4000 horsepower, and Rochdale, where six ejectors condense steam for 3000 horsepower. At a test recently made at the corporation of Hull Electricity Works, it was found that the actual power absorbed by the condenser was less than 2 per cent. of the total power, a very low figure indicating a very good efficiency of the ejector condenser.

TABLE 2. TEST DATA OF MULTIJET CONDENSER.

Inlet Temperature. Deg. F.	Outlet Temperature Deg. F.	Vacuum, Barometer at 30 Inches.
50	60	28.75
50	64	28.6
50	67	28.4
50	69	28.32
50	70	28.22
50	71	28.17
50	72	28.12
50	73	28.0
50	74	27.88
50	75	27.82
50	76	27.76
50	78	27.7
50	79	27.6

In order to prevent the possibility which exists in some forms of jet condensers of water getting back into the turbine or engine, the older form of Kőrting ejectors are provided with balanced nonreturn valves which effectually prevent any back flow of water from the discharge hotwell or sump, and automatic exhaust relief valves for the alternative branches to atmosphere. With the multijet ejector condenser an automatic vacuum breaker is provided in place of the nonreturn valve, as owing to the large diameter of the exhaust connections the cost of such a valve would reach a high figure. Incidental to this advantage is the reduction of loss of vacuum through friction in the exhaust pipes.

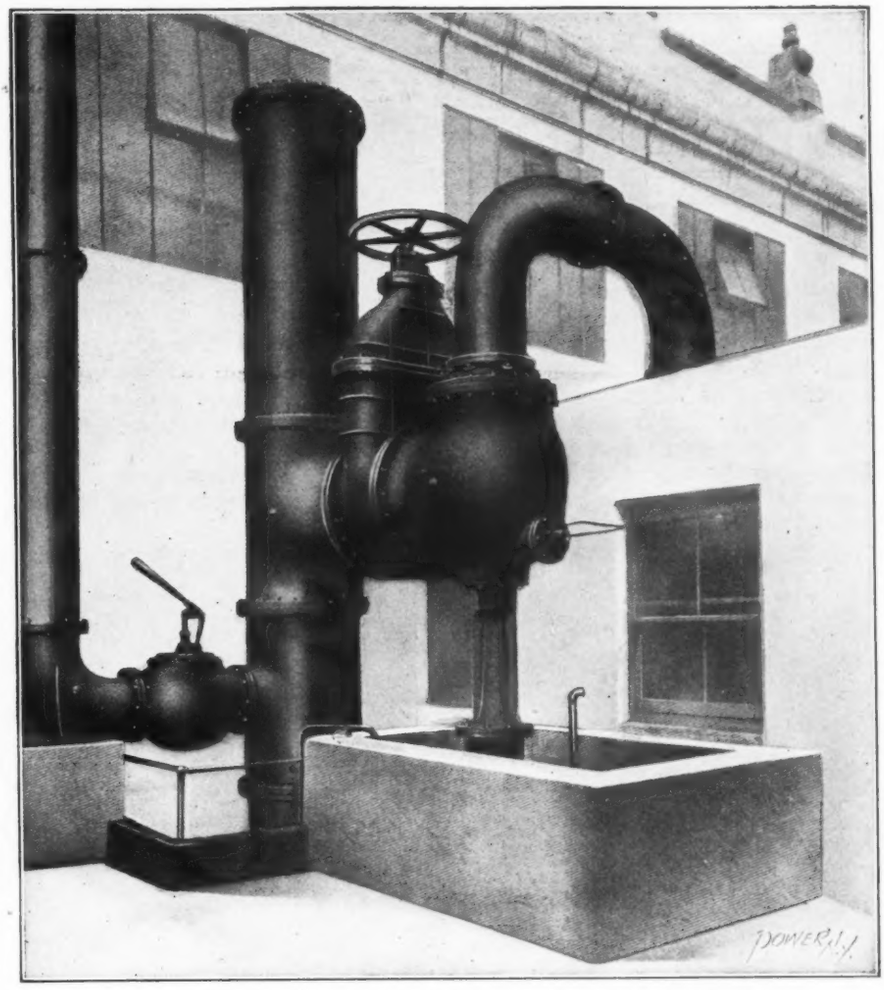


FIG. 12. KOERTING MULTIJET CONDENSER INSTALLED AT THE FRANCO-BRITISH EXPOSITION

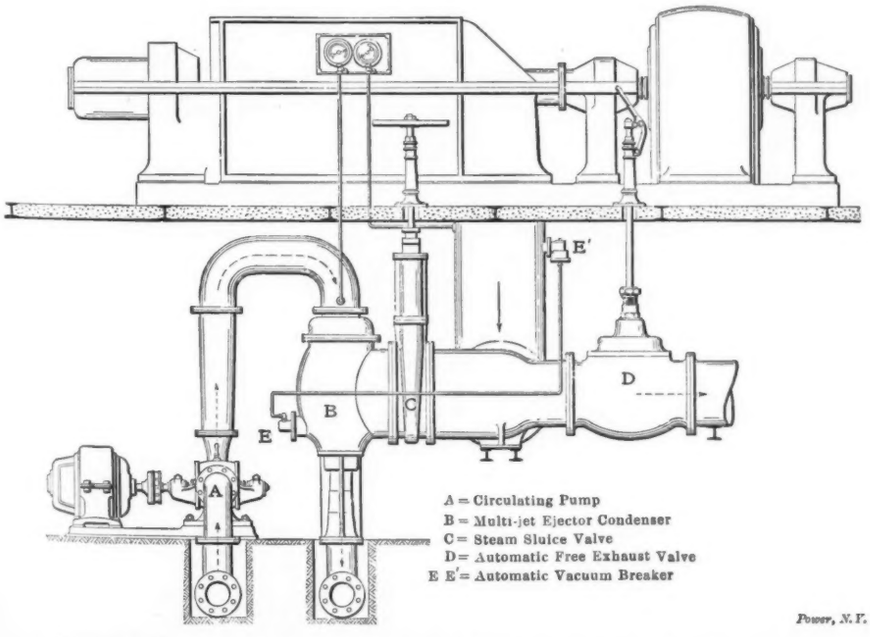


FIG. 13. KOERTING MULTIJET EJECTOR CONDENSER—ON STEAM TURBINE

The Board of Associated Engineers of Brooklyn and Queens has compiled some interesting statistics for the year 1909 in regard to the number and class of engineers, and boilers in Greater New York, from which we glean that there are a total of 12,583 licensed engineers in the greater city, divided as follows: First-class, 1623; second-class, 2582; third-class, 7751; public school, 136; fire department, 480. There were also issued 1342 firemen's certificates. During the year 723 new applications were received for engineers' certificates, 522 of which passed on the first examination, 56 on the second and 12 on the third.

There are 13,411 stationary boilers in the greater city, 2561 portable, 179 rollers, 229 pile drivers, 74 dredges, 89 derricks, 119 scows, 25 lighters, 255 barges, five schooners, five elevators and seven steam carriages requiring certificates, making a grand total of 17,464 boilers. Of this number 14,606 boilers were tested by the boiler-inspection department, 2354 being out of commission. The total amount of cash collected for boiler inspections was $30,172.

ELECTRICAL DEPARTMENT

ESPECIALLY CONDUCTED *to* BE *of* INTEREST *and* SERVICE *to the* MEN *in* CHARGE *of the* ELECTRICAL EQUIPMENT

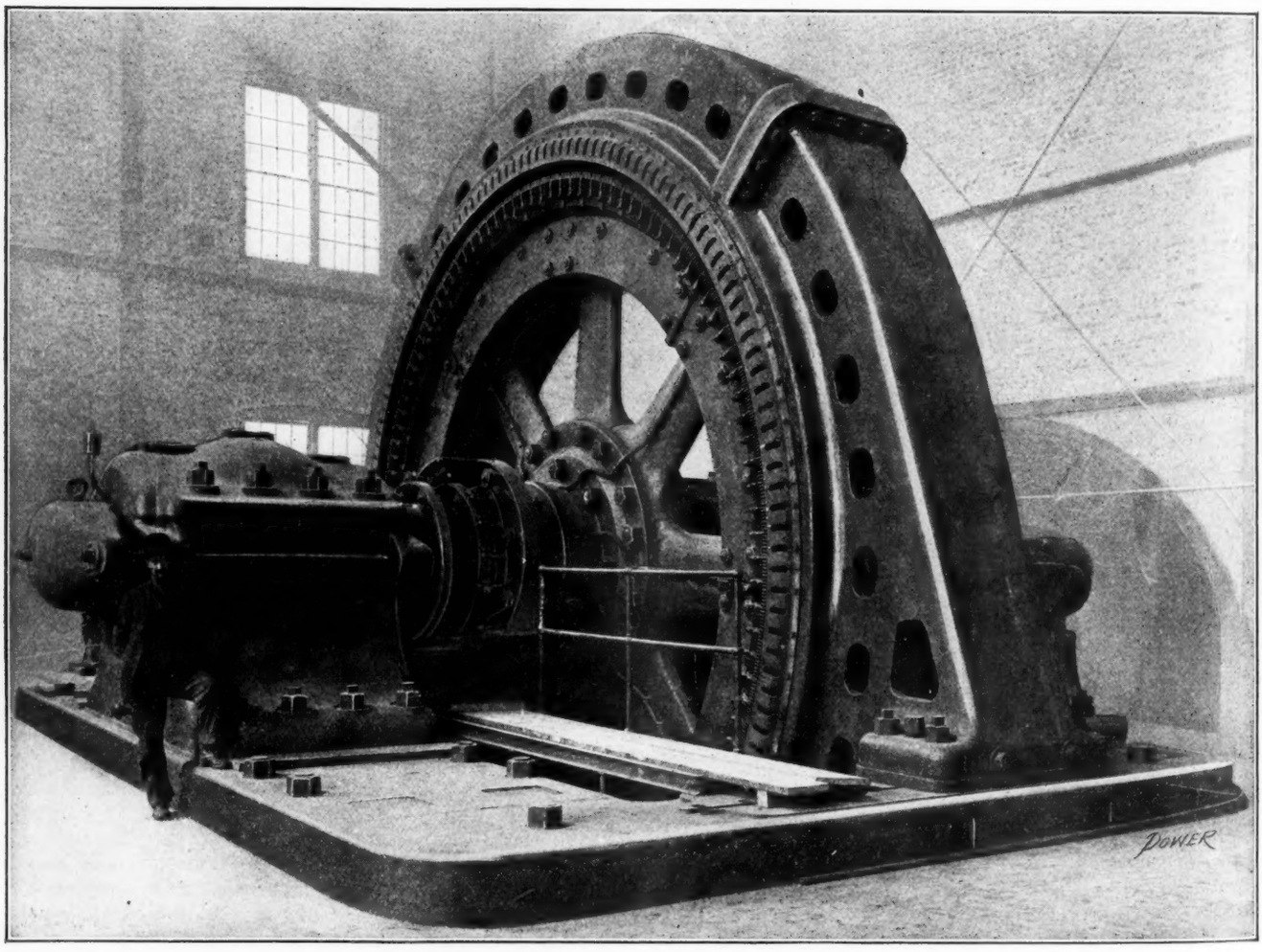
A Giant Motor

In the rail-mill department of the big Gary steel works there are motors aggregating about 30,000 horsepower, driving the rolls and other machinery. At first thought this figure would suggest a veritable litter of machines, but a glance at the accompanying picture will suffice to dispel that impression; it would require only five of these monsters to deliver the whole power output mentioned. There are only three of these, however, of 6000 horsepower each, two 2000-horsepower motors and several smaller machines.

The 6000-horsepower motor here shown is an alternating-current General Electric machine of the induction type, and is supplied with three-phase current at 6600 volts and 25 cycles. The main frame is 28 feet in diameter and the rotor is 21 feet in diameter. The complete machine weighs 392 tons. It drives a three-high five-pass mill, and has a maximum torque capacity equivalent to 16,500 horsepower.

The three 6000-horsepower machines run at different speeds, to suit the rolling-mill requirements. After the billets have been reduced to slabs about 15½ feet long, 14½ inches wide and 11½ inches thick, they are fed to the mill driven by one of these giants having 40 poles and therefore running at 75 revolutions per minute; the thinned-out slabs then go to the mill driven by the second of the three big motors, having 36 poles and running at 83 revolutions per minute, with a break-down torque equivalent to 18,700 horsepower, and finally to the mill driven by the third machine, which has 34 poles, runs at 88 revolutions per minute and has a break-down torque equivalent to 20,600 horsepower. Each of the three motors has a heavy fly-wheel incorporated in the spider of the rotor, to take the shock at "biting" the billets.

These enormous machines were too large to ship complete, and therefore had to be assembled in the Gary works by expert core builders and winders sent from the factory. Notwithstanding this disadvantage, the motors were started up and immediately put into commission without a hitch, excepting a partial short-circuit of one of the starting-resistance grids by a piece of arc-lamp carbon, carelessly dropped into the nest of grids by a trimmer.



A 6000-HORSEPOWER MOTOR IN THE GARY STEEL WORKS

Simple Ground Detectors for Power and Lighting Circuits

By W. T. Ryan

A very good and easily made ground detector for constant-potential direct-current circuits is represented diagrammatically in Fig. 1, where *L* and *L'* are lamps in series and connected across the terminals of the machine *G*. The lamps will burn with equal brilliancy and should be able to withstand double the voltage of the generator. If the line wire *A* is grounded, upon closing the switch *S* a current will flow through the galvanometer *g* and the lamp *L* will burn with greater brilliancy than *L'*. If the wire *B* is grounded the lamp *L'* will be the more brilliant. By using the galvanometer *g*, a very small leak can be detected. The galvanometer may be omitted, however,

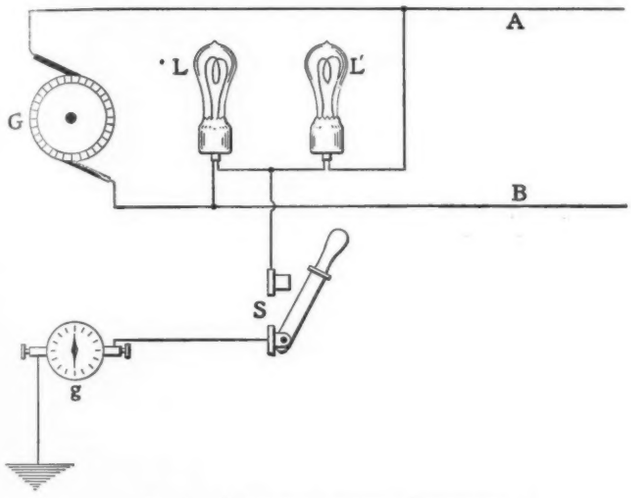


Fig. 1. A Simple Ground Detector

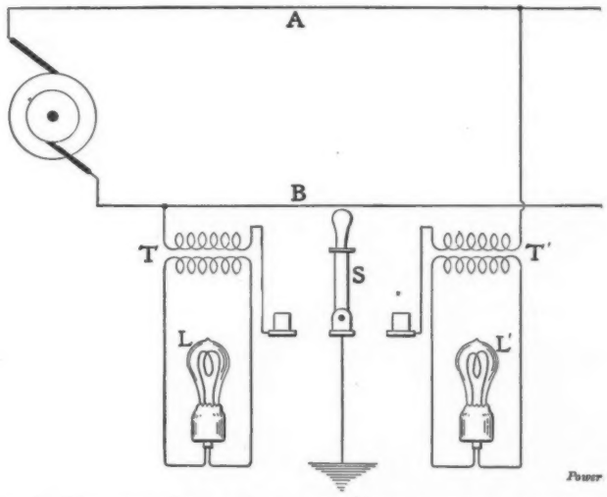


Fig. 2. Modification of Fig. 1 for High-tension Circuits

in which case the difference in brilliancy of the two lamps must be depended upon to detect the ground. If the switch *S* is also omitted the indicator becomes continuous and automatic but this is not advisable as the permanent ground connection makes it possible for a current to pass through anyone touching either side of the line.

Fig. 2 is a diagram of a ground detector for alternating-current circuits, which is similar to the one illustrated in Fig. 1. The difference between them is that the lamps take current through two small transformers instead of being connected directly to the line. This is necessary because the line voltage is usually much higher than the voltage for which the lamps are made. If there is a ground on the line wire *A*, connecting the switch *S* to the transformer *T* will cause the lamp *L* to light; correspondingly, if there is a ground on *B*, connecting the switch *S* to the transformer *T'* will make the lamp *L'* light up.

A better method and one which gives continuous and automatic indications of the condition of the line as to grounds is shown in Fig. 3, where *C* and *C'* are small condensers and *R* is a telephone receiver (an electric bell with a polarized coil may be used). The switches *S* and *S'* are kept closed. If either line wire becomes grounded the telephone sets up a hum which continues as long as the ground remains, and the connections are unchanged. If on breaking the circuit at *S* the hum still continues, the line wire *B* is grounded; if not, the ground is on the wire *A*.

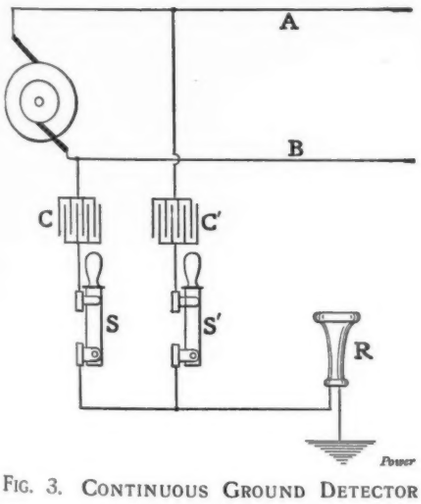


Fig. 3. Continuous Ground Detector

Fig. 4 is a diagram of the Bentley ground detector for railway and power circuits. A telephone receiver *R* is connected to a coil of wire *C*. To make a test, the direct-current generator *G* is shut down and the switch *S* closed. An alternator *A* is used to send an alternating current through the line wires, which are connected in parallel by the switch *S*. When the coil *C* is applied to either of the line wires, the telephone will hum, unless the coil happens to be applied at the point where the wire is grounded, as at *X*; when that point is reached the hum will cease. This arrangement may be used on alternating-current circuits without any change in generator connections. It is also useful in tracing circuits behind plastering or under floors, as in a residence.

Fig. 5 illustrates a method of locating grounds on arc-lighting circuits. At *L* is a bank of incandescent lamps connected in series, with one terminal of the group grounded. The lamps are of a voltage which is a multiple of the average drop between successive arc lamps in the circuit, while their total resistance is such that all just come up to normal candlepower when the full potential of the lighting circuit is applied to them. A dial switch, like that of a dynamo field rheostat, is connected up so

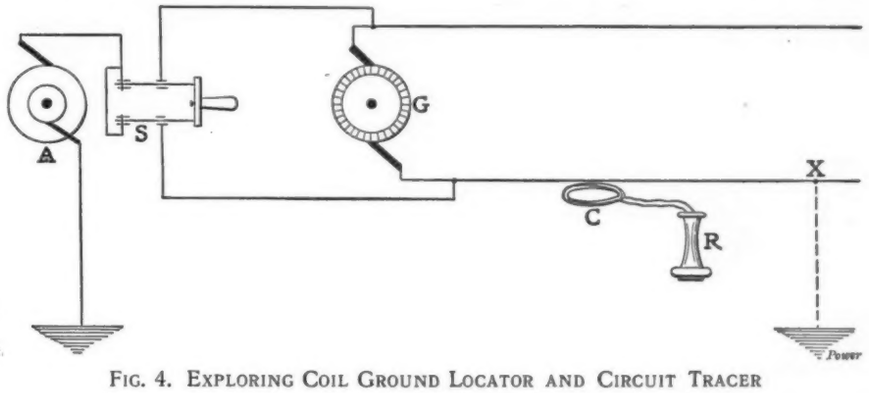


Fig. 4. Exploring Coil Ground Locator and Circuit Tracer

that the lamps can be cut out or in, one at a time. In testing for a ground, the switch lever is first connected to one side of the circuit and adjusted until the lamps not cut out just come up to normal candlepower. The switch is disconnected from *A* and connected to the other side *B* of the circuit and again adjusted until the lamps not cut out just come up to normal candlepower. If the sum

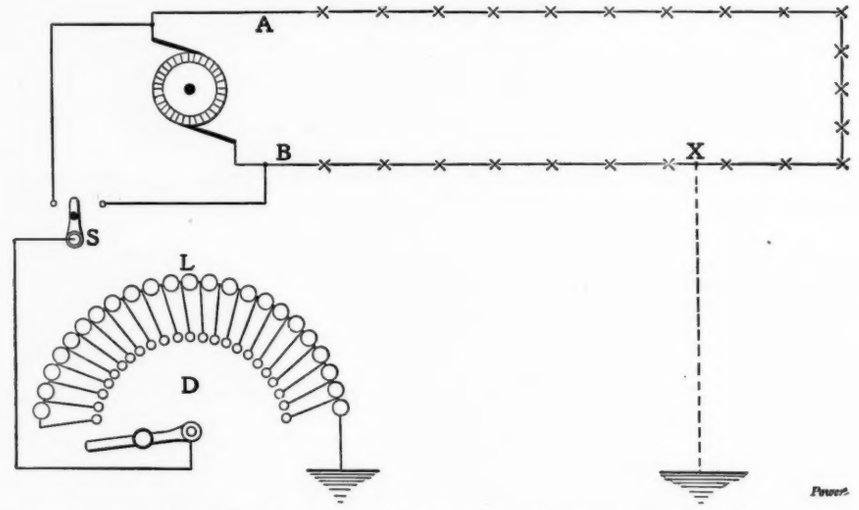


FIG. 5. STARKE'S DETECTOR FOR ARC-LIGHTING CIRCUITS

of the lamps lighted up in the two tests equals the total number of lamps in the bank, the arc circuit is dead grounded at a point *X* which is so located that dividing the number of lamps lighted in the first test by the number lighted in the second will give the same result as dividing the number of arc lamps between *X* and *A* by the number between *X* and *B*. If the number of incandescent lamps is the same as the number of arc lamps, no calculation is necessary; the number of lamps lighted in the first test will be equal to the number of arc lamps between *A* and *X*, provided the ground is a "solid" one.

Fig. 7 is a diagram of the Rudd automatic ground alarm for arc lighting circuits. Two condensers *C* and *C'* are connected together and across the dynamo terminals, and a connection is carried from between them to the ground through a magnet coil *M*. With conditions normal on the line no current passes through the magnet, but if a ground occurs the sudden change in voltage sends a momentary current impulse through the magnet *M*, energizing it and causing it to attract its armature. This pulls the latch out from under the lever *S* which falls on the contact below it and completes a circuit through the battery and bell shown. The bell rings until someone resets the latch and lever. The ground may then be located by the former or some other method.

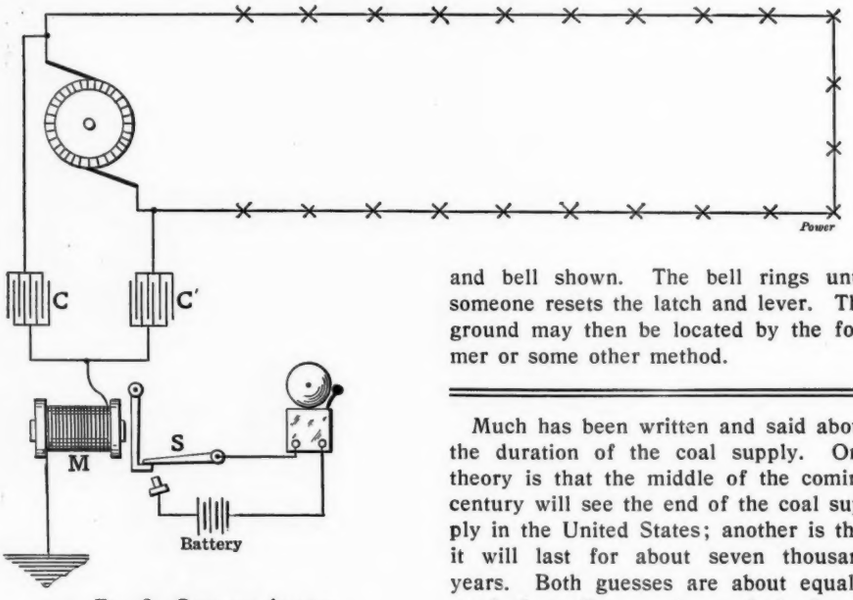


FIG. 6. GROUND ALARM

Much has been written and said about the duration of the coal supply. One theory is that the middle of the coming century will see the end of the coal supply in the United States; another is that it will last for about seven thousand years. Both guesses are about equally good, depending on the method of estimate. On the present increasing basis of production it would not take much over a century to utilize all the coal in sight in the United States. If, however, the annual production continues as now without increase the coal supply might last about seventy centuries.

CORRESPONDENCE

Mr. Saeger's Problem in Storage Battery Connections

The accompanying diagram (Fig. 1) meets the requirements of M. C. Saeger, stated on page 352 of the February 22 number. My diagram indicates a double-throw switch *A* and a single-throw switch *B*, both double-pole, but a four-pole double-throw switch could be used, of course. The switch points *M* represent merely a main switch for stopping and starting the motor, aside from Mr. Saeger's problem.

With the switches *A* and *B* closed upward, the former connects the line to one end of each half of the battery through lamps and the latter connects

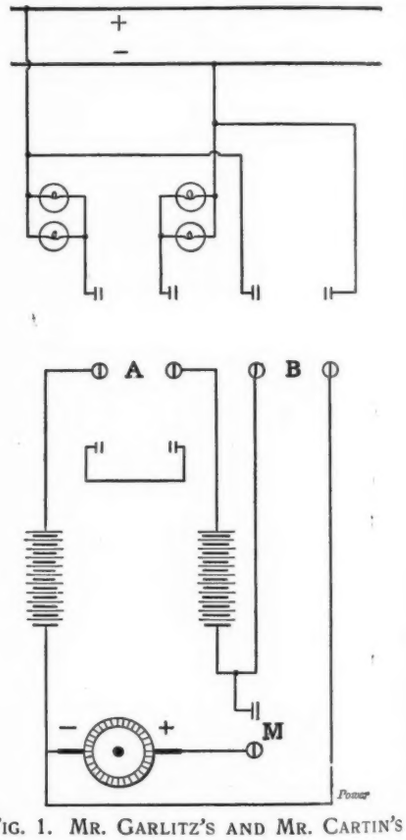


FIG. 1. MR. GARLITZ'S AND MR. CARTIN'S SOLUTION

the line to the other battery terminals direct and also to the motor, the switch *M* being closed. Opening *B* and closing *A* downward puts the battery and motor in closed circuit, disconnected from the line.

WILLIAM T. GARLITZ.

McKees Rocks, Penn.

[Exactly the same solution was received from James D. Cartin, Watertown, N. Y., except that Mr. Cartin showed no individual motor switch at *M*. This, however, is not strictly involved in the problem.—EDITOR.]

The accompanying diagram (Fig. 2) shows a method of simplifying Mr.

Saeger's storage-battery connections, illustrated in the February 22 number. It requires a four-bladed double-throw knife switch; when this is closed in the upper position the connections are such that the 50 storage cells are connected to the busbars in two groups of 25 cells each, in parallel with each other and in series with the lamps; the motor is then running directly on the busbar current.

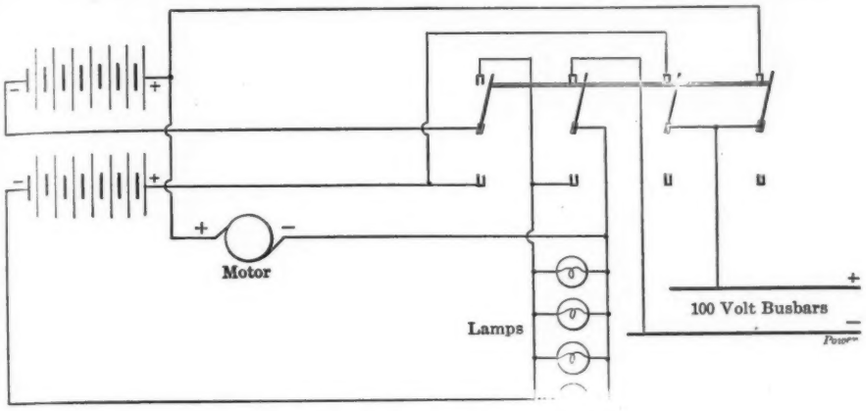


FIG. 2. MR. BICKLEY'S SOLUTION

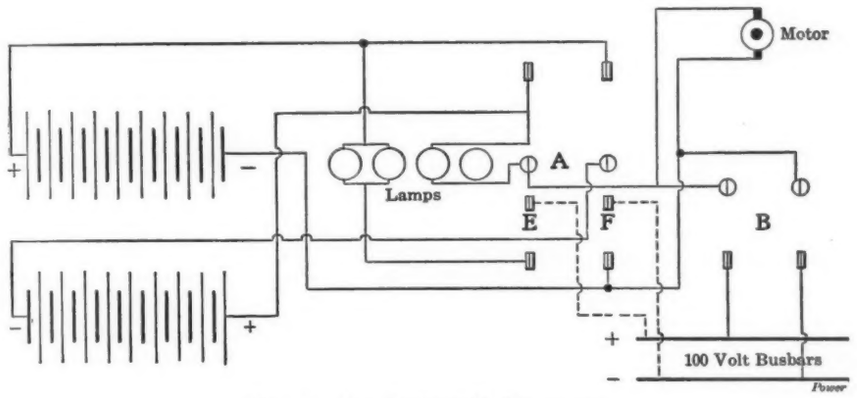


FIG. 3. MR. SPRAGUE'S SOLUTION

By closing the switch in the lower position the entire 50 cells are placed in series and connected with the motor only, the lamps being cut out and the whole system disconnected from the busbars. It is impossible to make the wrong connections with this arrangement.

EVERETT H. BICKLEY.

Pittsburg, Penn.

The arrangement shown by the accompanying diagram (Fig. 3) requires less switches than Mr. Saeger's method, and can be more readily operated. In the diagram, *A* represents a standard double-pole double-throw switch and *B* a double-pole single-throw switch. When the double-throw switch *A* is down and the single-throw switch *B* is closed, the two sections of the battery are in multiple, charging through their lamps, and the motor is running from the 100-volt source. When the battery is fully charged, the double-throw switch is closed upward and the other switch opened; this disconnects the battery from the busbars and puts it in series, connected to the motor. The switch *B* may be dispensed with by adding two clips *E*,*F*, to the double-throw switch as shown, and making connections from the busbars to these terminals as indicated by dotted lines. This arrangement would require but one switch, and the only objection would be a short interruption in the circuit while throwing over. If this is objectionable, the switch *B* must be used; leaving it closed until after the double-throw switch is thrown up, and then opening it, will make the change without stopping the motor.

G. S. SPRAGUE.

Geneva, Neb.

Possibly Excessive Line Drop

About three months ago the firm I am working for bought a 1000-volt alternating-current generator which was supposed to be a 17½-kilowatt 60-cycle machine, but which proved to be the field magnet of a 1000-volt 140-cycle 37½-kilowatt generator with the armature of a 2300-volt generator put in it and run at the speed necessary to give 60 cycles.

About 7000 feet from the generator is located a 15-horsepower Wagner motor which will not start and pull over eight horsepower. The line consists of one No. 6 and one No. 7 wire, terminating at a 1000-200-volt 15-kilowatt transformer, about 30 feet from the motor. When the load is increased to more than eight horsepower, the motor drops its brushes and blows an 80-ampere fuse.

If any readers of POWER can give me advice as to what the trouble is, I would like to hear from them.

LEROY WEEKS.

Auburn, Ia.

The Care of Rotary Converter Brushes

I have charge of four 750-kilowatt rotary converters, the commutators of which have not had a piece of sandpaper or stone on them for more than four years, and they are in the best condition of any that I ever saw; this is not pride or conceit—everybody who sees them says the same thing. I have found that the best way to set the brushes is so that one positive and one negative brush track with one another on the outside end of the commutator and one positive and one negative brush track with one another on the inside end of the commutator; that is, half of the brushes are set so that they touch the commutator where the other half do not touch. This, of course, reduces the wear to just one-half what it would be if the brushes on each stud tracked with all of the brushes on all of the other studs. In five years and four months I have only used 200 brushes, and all the brushes are in good shape now; as there are 111 brushes on each converter, I don't think that is very bad. On the alternating-current side of the converter the main thing is to keep it clean. I find that when I use No. 1 Albany grease on the rings the brushes do not cut at all, it is not so hard to keep things clean and the rings will wear a long while without having to be turned down.

W. H. RICHARDS,
Chief Operating Engineer.

Brockton, Mass.

The total rainfall, including snow and that on water areas, has been given by Dr. M. J. McGee, secretary of the Inland Waterways Commission, as 215,000,000,000,000 cubic feet a year. Half or more is evaporated. About one-third flows into the sea. The remaining one-sixth is either consumed or absorbed. Of the 70,000,000,000,000 cubic feet flowing annually into the sea less than 5 per cent. is used for power. It is estimated that from 85 to 95 per cent. of the volume is wasted in freshets or destructive floods. The theoretical power of the streams is over 230,000,000 horsepower. The amount now in use is 525,000 horsepower. The amount available, according to this report, at a cost comparable to that of steam installation is estimated at 37,000,000 horsepower.

GAS POWER DEPARTMENT

EVERY THING WORTH WHILE *in the* GAS ENGINE *and* PRODUCER INDUSTRY WILL BE TREATED HERE *for the* USE *of* PRACTICAL MEN

Splash Lubrication for Gas Engines

BY FRANK E. BOOTH

Splash lubrication is used on the majority of vertical gas engines, and this phase of the subject is especially interesting to me just now. The hot trunk pistons of splash-lubricated engines are directly responsible for considerable waste of oil, particularly in the case of high-speed engines. The oil is deposited on the inner surfaces of the pistons, where some of it is carbonized and so wasted.

On examining the contents of the inclosed crank chamber of a splash-lubricated engine which had been running for some time, I have found numerous small pieces of carbon, somewhat resembling tiny pieces of coke and feeling like cinders to the touch; this of course represents so much lubricating oil lost. Funnels with screens over the large ends are screwed into the caps of the inside bearings in order to keep these oil cinders from getting into the crank-shaft boxes.

If these engines have auxiliary exhaust ports the amount of lubricating oil used will be increased. The edges of the ports act somewhat like a scraper, removing oil from the surface of the piston; if the edges of the ports are chamfered, the loss will be reduced. The higher the level of the oil in the crank case the greater will be the oil consumption also, especially if the cylinders contain auxiliary exhaust ports. A decrease in oil consumption may be obtained, therefore, by keeping the oil level as low as possible without producing insufficient lubrication, which can be determined by a little experimenting. In order to operate with the minimum hight of oil in the crank case of a multi-cylinder engine, it is necessary that all rods dip an equal depth into the oil. As a rule the connecting-rod bolts strike the oil, as they project below the brasses; these may be cut to proper length or pieces of sheet iron may be secured between the lock nuts so as to project downward the proper distance.

It is the usual practice to put enough oil in the crank case to supply lubrication for the whole run; a better plan is to start the run with the oil level at its lowest successful working point and keep it there by feeding oil through a sight-feed lubricator on the top of the base. I have been able to cut the oil consumption in half by finding the proper level and keeping it uniform by means of a sight-feed lubricator.

Considerable oil is sometimes necessary to fill the crank case in the first place, resulting in waste of oil when the case is cleaned out. A good way to prevent this is to fill the case up with water to a level several inches below the striking parts of the rods, then make up to the proper depth with oil.

When the piston is moving downward it compresses the air in the crank case; if the boxes and the seams of the crank case are not tight, smoke from the hot oil on the inside of the pistons is sometimes forced out into the engine room. This is not only unsightly but extremely annoying and productive of much discomfort; it may be prevented by mounting a check valve in the top of the crank case and carrying a pipe or hose from the valve to the air-inlet pipe or outside the building.

The splash-lubrication system is simple, requiring but little attention, but the little experience that I have had indicates that this simplification may be obtained at the expense of unnecessary consumption of lubricating oil.

An Improved Water Pressure Gage

BY CHARLES C. SAMPSON

In rearranging the indicating and recording instruments in a blast-furnace gas-cleaning plant, it was necessary to overcome the frequent blowing out of the liquid in a U-tube water-pressure gage. This gage is used to show the pressure of the gas as it enters the scrubbers, and on account of sudden changes in pressure when the furnace slips, frequently goes for a few seconds as high as 40 inches of water.

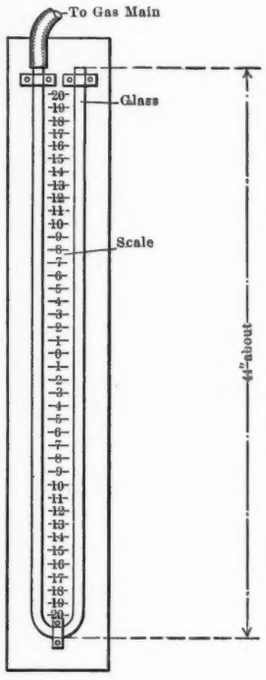


FIG. 1. OLD GAGE

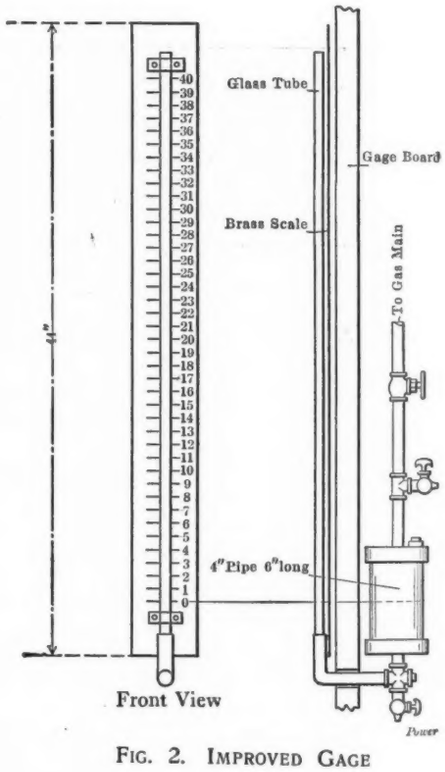


FIG. 2. IMPROVED GAGE

The old gage was a simple U-tube of ¼-inch glass, as shown in the sketch, Fig. 1. The quick rise in pressure would get the water moving so fast that it would blow out and then the gas would have to be shut off the gage and the latter carefully refilled to the zero mark.

A gage (shown in Fig. 2) was made consisting of a single ⅛-inch glass tube with a rubber connection through the gage board to a piece of pipe, 4 inches in diameter and 6 inches long, on the back of the gage board. The pet cocks shown are for adjusting the gage to zero.

To prepare the gage for use, the pet cocks and valve are closed and the 4-inch pipe is filled with water; then the lower pet cock is opened until enough water runs out to allow the water in the glass tube to stand at zero. The filling plug is then replaced and the valve in the pipe leading to the gas main is opened. To test the gage for zero, close the valve and open the upper pet cock.

On account of the lowering of the surface of the water in the larger pipe the scale of 40 graduations should be only

$$\frac{40 \times (4^2 - \frac{1}{8}^2)}{4^2} = 39.96$$

inches long, but for ordinary work this error could be neglected.

The large body of water in the 4-inch pipe "chokes" the tendency toward a sudden rush of water through the tube and with the compression space above the water makes it almost impossible to blow out any water under working conditions.

While the same cause of pressure fluctuations does not exist in power producer plants, this gage will be found useful for damping out the violent fluctuations which occur from other causes.

An Obscure Ignition Trouble

By W. R. Willard

Why does a gas or gasolene engine sometimes refuse to work when everything seems to be in first-class condition? This is the question which bothered me for some time not long ago, and I found

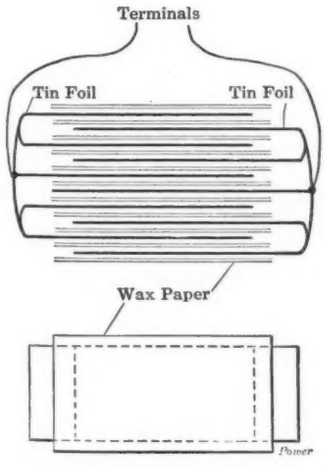


Fig. 1. Condenser Structure

the answer simple—after I did find it. I have a two-cylinder 20-horsepower engine which proved a puzzle to several experts. When it ran slowly, with no load, it was all right, but as soon as the speed was increased and the load put on it would miss fire, sometimes in one cylinder and sometimes in the other. I inspected every part without results, and got two or three engine experts at it, but they gave it up. One advised me to get an electrical expert to look at the coil and I did; he said it was in first-class shape. It is a jump-spark system, and all the electrical parts were, as far as could be seen, working well.

Finally I took half a day off, with the determination to make it work or else to

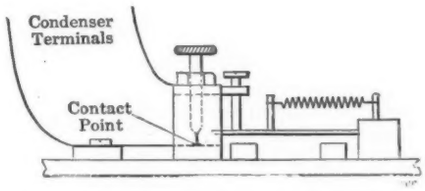


Fig. 2. Condenser and Vibrator Connections

throw it out. After a long hunt I found that there was a considerable spark at the contact points of the vibrator, and no adjustment would lessen it. The platinum was wasting away rapidly. When the engine ran slowly the contacts would keep cool, but when it was speeded up the points got so hot that they stuck together, and as soon as this occurred, of course, the engine missed fire. The coil was a self-contained one, with the condenser inside, and this is where the trouble was. The coil was a French one and no one kept the parts nor would anyone attempt to put a new condenser in it. After considerable experimenting with it I succeeded in getting one that worked fine. For the benefit of those who may have had the same trouble I will describe the construction of it.

A condenser consists of two sets of tinfoil sheets which are separated from each other by sheets of wax paper, as shown in Fig. 1. About ten sheets of tin foil 12x18 inches and twenty-five sheets of wax paper of the same length but one inch wider will be sufficient for the average spark coil. Two sheets of wax paper will be required between each of the sheets of tin foil to prevent the pressure from breaking through. Care must be taken to prevent the tin foil from touching at the edges, and after each sheet of paper has been put in place it should be pressed with a warm iron. The iron should not be hotter than 200 degrees Fahrenheit; a higher temperature will spoil the paper. If it is desired to fold it this may be done while it is still warm. The closer the sheets of tin foil are together the better, so long as no leakage is incurred. All wrinkles must be pressed out and the surfaces must touch everywhere. The connecting up of the condenser to the coil is very simple, as shown in Fig. 2. If the condenser is properly made there will be no spark at the contact breaker to speak of.

In some cases, where the battery gives more than six volts, it will be necessary to use twice the number of sheets of tinfoil and put three sheets of paper between each sheet. The best number can be found easily enough by a little experimenting.

A Bad Case of Tar

About four months ago there was put in operation in our plant a three-cylinder, vertical 225-horsepower gas engine and suction producer. Everything had been giving excellent service for two months, when on one Monday morning, after the plant had been running about an hour, the attendant noticed that the engine was pulling harder than usual. Looking the engine over and finding no trouble he went out to the producer and there found the tar extractor standing still, with the belt only slipping around the pulley. Catching the shaft with a pipe wrench, he gave the extractor a start and it went right up to normal speed. But within less than three minutes after that the engine stopped. After having pumped up the air to 200 pounds we tried to start again as usual, but the engine would not move; so we tried "barring" the flywheel over with long levers, but the air and the levers combined couldn't start it.

The plant contains a 150-horsepower Corliss engine which can be connected to the same shaft as the gas engine by a clutch, so we connected it up, turned air of 200 pounds pressure into the gas engine and steam of 145 pounds pressure into the Corliss engine, but these combined could not move it one bit.

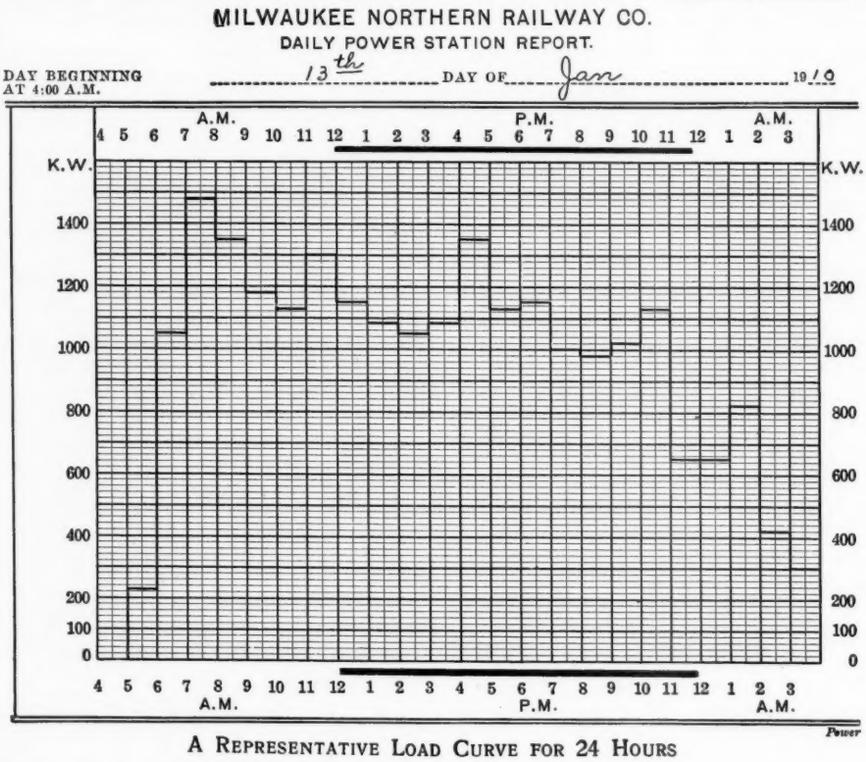
Then we dismantled the cylinders and found them simply clogged with tar. We cleaned out all the tar we could and then made a mixture of mineral oil and kerosene, poured about four gallons of this into each cylinder and let it stand over night. The first thing next morning we turned live steam into the water jackets, and after the engine had become thoroughly warm we were able to draw the pistons out with a chain hoist. The rings were so badly stuck with tar that blow torches had to be used to burn the tar out before they could be removed from the pistons. In the gas pipe from the extractor and in the inlet manifold we found large quantities of tar.

After having thoroughly cleaned all the parts and assembled the engine it started promptly and ran smoothly. We have had no further trouble except having to clean out the inlet manifold occasionally.

This trouble happened last December, but we have not had any difficulty in preventing it from happening again, although our means of extracting the tar does not seem to be as effective as it should be.

Performance of the Port Washington Gas Power Plant

One of the first large gas-power generating stations to be built in this country for traction service only was completed in the fall of 1907 by the Milwaukee Northern Railway, and has been in service since that time. In the operation of the power plant, which now supplies only 54 miles of what will be eventually a 112-mile traction line, it is found that one generating unit will carry the load most of the time. However, the load sometimes exceeds the capacity of one machine, particularly during stormy weather, and a second is then put in service. If the total load does not exceed 2000 kilowatts, one engine is operated on one side only, that is, the gas is cut off from one set of tandem cylinders and the generator is driven by a single tandem gas engine, which also drags the side not in operation. This, of course, is a most severe operating condition, but no serious difficulty has been experienced in securing and maintaining parallel operation.



A Representative Load Curve for 24 Hours

The accompanying daily power-station report for January 13, 1910, shows the performance of the station under the conditions just mentioned, with the additional circumstance that a heavy snow storm was raging throughout the period covered, making traffic exceedingly difficult. On that day, from 6:30 a.m. until 11:15 p.m., engine No. 2 was run in the regular manner and engine No. 1 with one side cut out, giving a total generating capacity of 2000 kilowatts. Generator No. 1 delivered 5125 kilowatt-hours and No. 2 generator, 17,525 kilowatt-hours, a total output of 22,650 kilowatt-hours. This gave an average load factor of 60 per cent. on engine No. 1 and 78 per cent. on engine No. 2. During the day 51,000 pounds of Pocahontas coal were used in the producers, amounting to 2.25 pounds of coal per kilowatt-hour of output at the switchboard. On the 24-hour run 12 gallons of gas-engine cylinder oil were required.

The power house is located in the city of Port Washington, Wis., directly on the harbor front. The equipment consists of Loomis-Pettibone down-draft producers, built by the Power and Mining Machinery Company, and Allis-Chalmers gas engines and alternating-current generators. The generating units are rated at 1000 kilowatts each, but this is a conservative rating.

LETTERS

Temporary Excess of Steam in a Producer

In my short letter published in the January 25 issue of Power regarding temporary excess of steam in a producer, I used the time-worn expression "take the bit in her teeth," never thinking that any engineer would not understand it. Probably I should have avoided slang and said that the addition of steam to the producer fire when the load comes on suddenly, will so enrich the gas that the engine will take up the added work without losing speed. Without the extra steam it will take several minutes for the producer to attain the rate of gasification demanded by the engine. Several times I have seen engines stall from this cause.

This additional steam increases the percentage of hydrogen in the gas and, of course, will lower the temperature of the fire if continuously supplied. The excess steam should be gradually diminished until only the usual amount is passing through the fire.

Mr. McArdell in the March 1 issue of Power says: "If the load is left off for, say, a half hour, the fire becomes deadened to such an extent that the gas is not good and the rate of gasification falls below a workable figure."

This is just the condition for which this remedy is prescribed. It is not surprising that when the load is taken off, and the draft through the fire thereby reduced, the temperature of the fire drops and less steam is drawn through.

It seems to me that there must be something wrong in the design of a producer in which any amount of poking will cause an excess of steam to be drawn into the fire.

As to the gas being "diluted" by excess steam, such a condition could hardly occur in a suction plant unless the steam was deliberately supplied from some outside source.

Dow R. Walker.

Lebanon, Penn.

Cooling Water Connections

I have charge of two vertical three-cylinder gas engines of different makes and the cooling-water conditions are as different as the engines themselves. The older and smaller engine has a very small water space around the cylinder and small openings from the cylinder to the head jackets, making the velocity of the water up through the cylinder jacket far too rapid and the cylinder too cold when the rate of water feed is sufficient to keep the head cool enough to prevent premature ignition. The inlet water connections of this engine are all connected together by a cast-iron manifold, with no way to control the water to each cylinder, and the outlet connections are arranged the same way; consequently the one nearest the intake end of the manifold runs the coldest when the water passage is clean and hottest when it is dirty, it filling up the quickest. This is one case where thermostat control would not do.

The other engine is of recent and more liberal design as to cooling-water space and the water feed to each cylinder is controlled by a valve, so the cylinders can be made to run all alike as to temperature. There is ample space in the jackets around the cylinder and on the heads, with large openings between, making the velocity low and the temperature more uniform throughout. No trouble is experienced with the engine if the discharge water is not allowed to go above 140 degrees, but at temperatures much above that the pistons show a tendency to heat and burn the oil.

J. A. Mawhinney.

Franklin, Penn.

FROM READERS *with* SOMETHING TO SAY

PRACTICAL INFORMATION *from the* MAN *on the* JOB: A LETTER GOOD ENOUGH *to* PRINT *is* GOOD ENOUGH *to* PAY *for*

Renewing Shaft Lining of Curtis Turbine

There was something decidedly wrong with the 2000-kilowatt Curtis turbine. The buckets rubbed even with medium loads, and the machine acted as if it was completely out of balance. As there was a slight jingling noise at the step bearing it was concluded that the step blocks at least needed renewal.

In Fig. 1 are roughly shown some of the details at the turbine shaft below the wheels. The first step taken was to block the machine up with bolts of proper size placed under the flexible coupling that unites the generator shaft to that of the turbine. Two long screws with a free running nut on each were then secured as shown at *A*, replacing two of the cap bolts that retain the heavy supporting plate in place. After arranging suitable blocking, a jack with the screw extended was placed under the casting and the rest of the cap bolts were then removed. While one man kept the nuts on the long screws a quarter of an inch in advance of the descending block, the other man manipulated the screw of the jack. This method lowers the supporting plate and the two step blocks which generally descend with it, much more quickly and evenly than the use of the long screws alone.

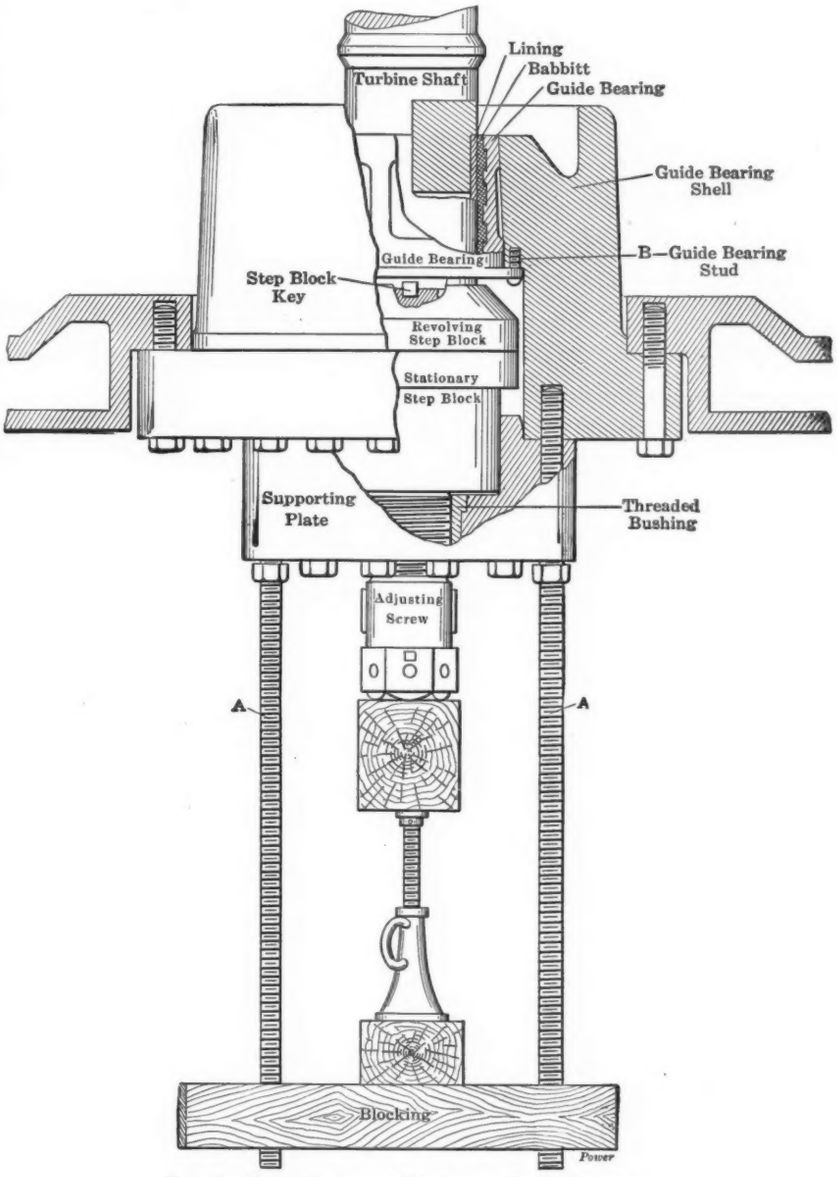


FIG. 1. PART SECTION THROUGH STEP BEARING

The cause of the jingling was found to be two guide-bearing studs *B* that had worked loose. After removing the revolving step block, it was decided to lower the guide bearing. Before backing off the remaining supporting nuts, a narrow piece of 2-inch plank *A*, Fig. 2, was cut long enough to fit under the flange of the bearing, and the jack,

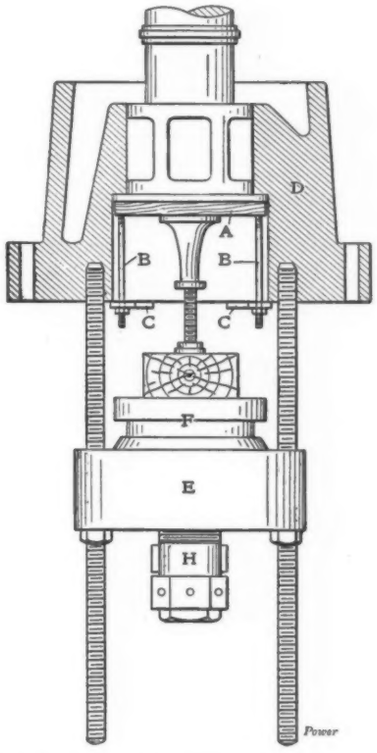


FIG. 2. LOWERING THE GUIDE BEARING

bell side up, with screw extended and resting on the stationary step block, was placed under it. The nuts being removed, two long studs *B*, Fig. 2, were screwed into tapped holes in the flange, resting on the step-bearing shell. These are supposed to be capable of loosening the bearing, but in the present case they were insufficient. Recourse was then made to ½-inch straps *C* placed crosswise of the opening, and the use of nuts on the drawing studs. This, after some effort, resulted in the bearing gently dropping onto the 2-inch plank, and the rest was easy. When examined, it was found that the water channels were

nearly plugged up, and that the babbitt was badly scored and worn. Passing the hand around the shaft lining it was discovered that this also had heavy scoring lines, and it was decided to renew it. This necessitated the lowering of the guide-bearing shell *D* as, from experience, it had been found cheaper to split the brass lining than to try to remove it by jack and hooks as suggested by some who evidently had not personally tried to remove this stubborn member. Before removing the supporting plate *E* from its position on the long screws, it is best to force out the step block *F* from its tightly fitting recess (see Fig. 1). This can be done either by means of the adjusting screw *H*, in which case the plate must be raised, and a few cap bolts replaced to resist the

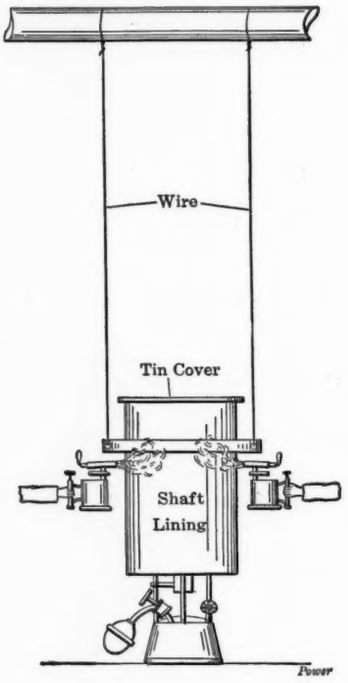


FIG. 3. HEATING NEW LINING WITH TORCHES

twisting strain necessary, or as follows, the way in which it was done in our case: The adjusting screw *H* was removed and carefully laid to one side to prevent injury to the thread. Then, after wedging two 4x4-inch blocks diametrically opposite between plate *E* and shell *D*, and balancing on top of the jack, placed directly under, another block of a size to pass through the vacant screw hole, the step was easily removed, when strain was applied to the jack. The guide-bearing shell *D* was lowered in precisely the same manner as was the supporting plate. When out of the way, a man entered the exhaust chamber with a suitable, keen but thick-nosed chisel, and cut a groove lengthwise of the lining. To make things a little more comfortable for him the air pump was set to running slowly. When the incisions were almost completed, the lining cracked along the line of the groove. As the chisel both cut and expanded the metal, it was easily removed. The new lining was heated with torches in the manner shown in Fig. 3. While heating, a ring of iron was turned in the lathe to the exact size of the turbine shaft, and was used as a gage to determine when the lining was sufficiently heated. When ready two men entered the exhaust chamber while two others quickly carried the lining and passed the wires to the former, and in a jiffy the thing was in place and supported while cooling by the step-block key (Fig. 1), under which was rigidly fitted a stout piece of planking.

In replacing the new guide bearing, square-shanked studs were used to fit into the square holes that had been cut in the flange at the manufacturer's shops. When all was reassembled, 4 inches was sawed off the baffler to give the bearing more water. An additional weight was added to those of the accumulator. Incidentally, the turbine is now given a rest as often and as long as it is possible to spare it.

H. HUGHES.

Roxbury, Mass.

Boiler Protection

It is a fact that boiler explosions are but too frequent. The results of an explosion are usually serious and very costly with respect to life and property. If the internal pressure did not exceed the strength of the boiler at any time it would be less dangerous than a motor. However, since our material is not always reliable and the work is often faulty and the engineer is sometimes neglectful we must be on the alert for indications which point to danger. If the boiler has been permanently weakened it is sometimes easier to reduce the pressure than to repair the weakness. A weakened boiler is a poor and unsafe piece of apparatus to have in use and

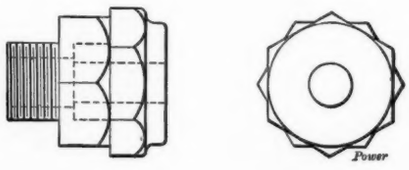


FIG. 1. FUSIBLE BOILER PLUG

the best thing to do, if it cannot be made as strong as ever, is to throw it out and put a new one in. I have met engineers who did not realize the seriousness of carelessness, probably due to the fact that they had never met with any serious accident. A reliable, careful engineer, regular inspection and a good boiler are the best safeguards against trouble, but in some cases even these have failed to avert accident.

Not long ago I was talking to an engineer from a neighboring town who was buying a new safety valve. I learned that he had had a mishap with his safety valve the day before. The pressure in his boiler had mounted to a point considerably above the blowoff point and the safety valve would not work. The situation was alarming. He had the presence of mind, he said, to seize a large hammer which was at hand and break the valve. In a few seconds the boiler was empty. Why it did not explode is the mystery. His presence of mind must have been absence of mind after all. If he had known that water at atmospheric pressure boils at 212 degrees Fahrenheit, and as the pressure is increased to, say, 90 pounds, the boiling point rises

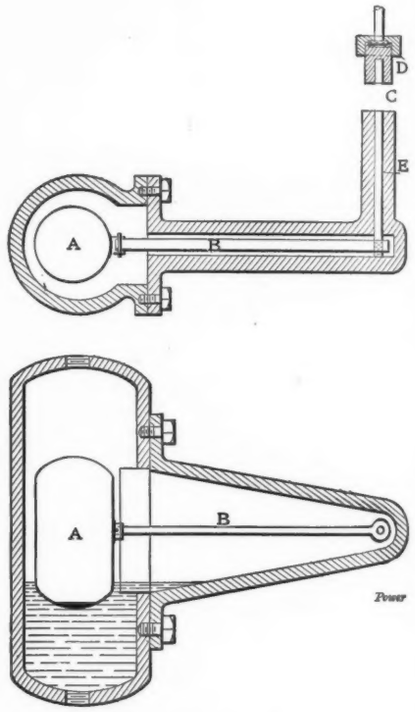


FIG. 2. FEED-WATER REGULATOR

to 312 degrees, and if the pressure reaches 150 pounds the boiling point will rise to 356 degrees, and that when the pressure is suddenly released a large quantity of water is transformed into steam and an explosion like that of gunpowder results, he would have used his scraper instead of his hammer. Where the pressure is very high the fire should not be drawn until some sand, salt, ashes or even fine coal has been thrown onto it to absorb part of the heat. Never put water on a fire under these conditions as steam will be formed which after being heated to a high temperature, will carry the heat directly to the boiler and an explosion will occur.

Boilers may give way under ordinary pressure as a result of poor construction, neglect or because of some change or

alteration in the fittings. If rust or sediment is allowed to accumulate in or on the plates or tubes the metal will burn and become weak. Fittings should not be changed or fixed by anyone who is not thoroughly experienced. Some time ago a firm in this city sent its engineer to ascertain why a gage which it had sold would not operate after it had been mounted. He found that a hole had been drilled through the reinforcing plate only and not through the boiler plate as it should have been. The man who had done the work explained that he was afraid to put a hole in the boiler because he thought it would not hold steam if he did. This was not a serious mistake as far as explosions are concerned but it goes to show what thoughtlessness may do.

Low water is another source of trouble to the engineer. If the water does not cover every part of the boiler which is exposed to the heat the boiler will burn and explode if the engineer does not happen to be at hand to act promptly. To guard against this, every boiler should be equipped with some device which will regulate the hight of the water or give a low-water signal. In some places the law compels every boiler to be equipped with fusible plugs similar to the one shown in Fig. 1. This is a safeguard against explosions but not against low water and its sphere of usefulness is limited. A device that will blow a whistle or ring a bell when the water is too low or too high is much to be preferred and saves much trouble. In some boiler rooms there are two whistles, one for high and one for low water. Still better than this is a device which automatically regulates the hight of the water in the boiler. Such an apparatus is shown in Fig. 2. It was designed by an engineer who found that the whistle gave much annoyance.

Its operation is very simple. The float *A* actuates the arm *B* which turns the small shaft *C*. This shaft is connected to the throttle-valve stem by the small clutch *D*. The throttle valve, not shown, is one which works very easily. The bearing *C* is steam tight and offers very little friction. After it has been connected to the boiler the throttle of the pump is regulated as closely as possible by means of the clutch. If the water should begin to go down, the float will also go down and open the throttle. This works as well when the water rises and keeps it at the same hight all the time. The float is of special design, being made of thin copper and inflated with air to keep the pressure of the steam from collapsing it. The case may be made of iron or brass as desired.

This device works well if it is carefully constructed but like every mechanism which is float operated it is not infallible and must receive careful attention. Any engineer, whether he has such a device or not, will find it always in order to watch the glass and keep awake.

W. R. WILLARD.

Toronto, Canada.

Governor Trouble

I am running an 18x30-inch Corliss engine. When starting up I have to keep the pressure shut off from the safety stop until the engine is up to speed. If I do not do this, after the engine has made five or six revolutions the governor will drop and stop the engine. The belt does not slip.

After the engine is up to speed the governor will pick up and work all right. The trouble is only in starting up.

I would be glad to have POWER readers tell me what is to be done under such circumstances.

W. J. MCMAHON.

Oconto Falls, Wis.

Examination in Thermodynamics

As an indication of the kind of work which is being done at the Polytechnic Institute of Brooklyn, as well as to give our readers an opportunity to test their own ability to solve such problems we publish below the examination in thermodynamics undergone by the junior class for the first semester of the present year.

EDITORS.

1. A steam engine receives steam containing 1100 B.t.u. per pound at 300 degrees Fahrenheit, discharging this steam at 212 degrees Fahrenheit. Find the minimum possible steam consumption per indicated horsepower-hour.

2. Find the volume of a tank to hold 10 pounds of air at 2000 pounds pressure per square inch, absolute, the temperature of the air being 65 degrees Fahrenheit.

3. Assume a cycle to be bounded by polytropic paths, 1—2, 2—3 and 1—3. We have given

$P_1 = P_2 = 100{,}000$ pounds per square foot,

$V_2 = V_3 = 40$ cubic feet,

$T_1 = 3000$ degrees Fahrenheit,

$P_1V_1 = P_3V_3$.

Find the amount of heat converted to work in the cycle, if the substance is one pound of air.

4. Given the formula for increase of internal energy between *a* and *b*,

$$\frac{P_bV_b - P_aV_a}{\gamma - 1}$$

find the *heat absorbed* along the path $PV^{1.7} = c$ if the substance is one pound of air, $P_a = 1000$, $V_a = 1$, $V_b = 2$.

5. The isodiabatics *ab*, *cd*, are intersected by lines of constant volume *ac*, *bd*. Prove

$$\frac{T_a}{T_c} = \frac{T_b}{T_d}$$

6. Compute for steam at 100 pounds absolute pressure, 95 per cent. dry ($T = 327.8$ degrees Fahrenheit, $H = 1186.3$, volume $= 4.429$, for dry steam) the values of *H*, *L*, volume, and total entropy from 32 degrees Fahrenheit for the wet steam.

7. Show on the NT plane the combined ideal diagrams of the air compressor and air engine and illustrate the economical effect of multistage operation.

8. Find the work expended per cycle in an air compressor without friction or clearance, compressing one cubic foot of air isothermally from 14.7 pounds per square inch pressure to a pressure of 100 pounds, both absolute. The intake air is at 32 degrees Fahrenheit. How would the amount of work compare with that necessary in an actual compressor?

9. (a) Show on the TV and TN diagrams the path of H_2O from liquid at 32 degrees Fahrenheit to dry steam at 300 degrees Fahrenheit, at constant pressure. (b) Explain

$$PV^{1.7} = c$$

$$H = H_{212} + a(t - 212) - b(t - 212)^2$$

$$e = \frac{P(W - V)}{778}$$

10. Find the factor of evaporation, in Problem 6, if the feed water is at 212 degrees Fahrenheit.

11. Steam 95 per cent. dry at 100 pounds pressure expands adiabatically to two pounds pressure, at which the entropy of the liquid $= 0.1749$, entropy gained during vaporization $= 1.7431$. Find its dryness after expansion. (See Problem 6.)

12. The expansion described in Problem 11 is a part of a Clausius cycle. At two pounds pressure, $T = 126$ degrees Fahrenheit. Find the efficiency of the cycle. (See Problem 6.)

EXPLANATORY NOTES

Polytropic paths are those along which the specific heat is constant. They conform to the general equation

$$PV^n = a\ constant$$

in which *n* may have any value from zero to infinity.

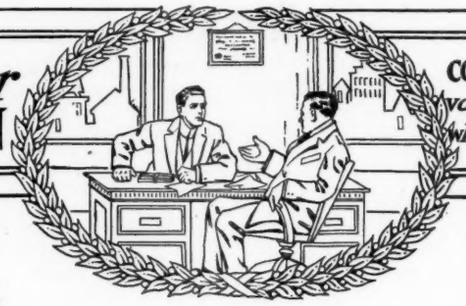
Isodiabatics are polytropic paths having the same values of *n*.

The *T V diagram* is one the coördinates of which are temperature and specific volume.

The *T N diagram* is one the coördinates of which are absolute temperature and entropy.

The *Clausius cycle* (sometimes called the Rankine cycle) is that of complete adiabatic expansion.

SUBJECTS *under* DISCUSSION



COMMENT, CRITICISM, *and* DEBATE *upon various* ARTICLES, LETTERS *and* EDITORIALS *which have* APPEARED *in* PREVIOUS ISSUES

Steam Piping in Power Stations

With reference to Perry West's article on "High-Pressure Steam Piping," which appeared in the issue of POWER of January 18, the unit system is, in my opinion, generally the best except for very small plants. With this system it is desirable that each engine unit shall be situated directly opposite the boiler or boilers which supply it with steam in order that the steam pipes normally in use may be as short as possible and in order that the station may be extended without upsetting the arrangement.

With high-capacity steam-turbine units which occupy little space in proportion to their power, different arrangements have been tried or proposed to keep the boiler room the same length as the turbine room without making the latter longer than is necessary to accommodate the machines.

One arrangement comprises two boiler rooms, one above the other, with a double row of boilers with a central, longitudinal, firing aisle in each boiler room. This arrangement has been adopted in several power stations in America but does not appear to have appealed to European engineers.

Another arrangement consists in placing the boilers back to back in short rows at right angles to the length of the engine room, as shown in Fig. 1. This arrangement was partly illustrated by Fig. 2 of Mr. West's article. If three boilers are not sufficient for each turbine, four or more may be placed in each row.

A third arrangement which has been proposed consists in providing one row of single-ended, and one row of double-ended, boilers, as shown in Fig. 2, the single-ended boilers being nearest to the engine room. There are, with this arrangement, two firing aisles.

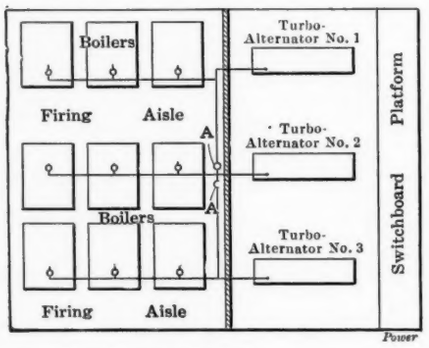


FIG. 1

Another arrangement is that shown in Fig. 3. According to this arrangement there are two longitudinal rows of single-ended boilers with a firing aisle between them. There being, in the arrangement shown, three boilers for each turbine, this arrangement is not strictly in accordance with the unit system, as any two units cannot at the same time be in full operation and yet completely isolated; but most of the advantages of the unit system are obtained. The arrangement shown in Fig. 1 is, however, in my opinion, preferable, as is also that shown in Fig. 2 if suitable boilers can be obtained.

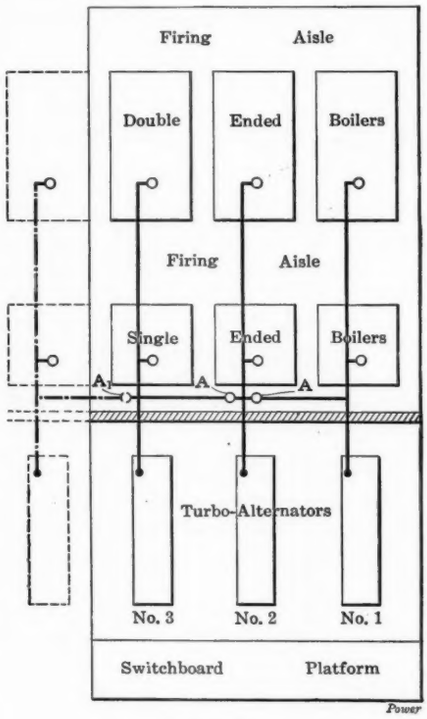


FIG. 2

The figures must be taken as only diagrammatic. The small circles on the steam pipes indicate valves. As regards the exact arrangement of the steam pipes, with bends, expansion joints, etc., opinions differ.

An advantage of the spider arrangement, mentioned by Mr. West, is that all the valves on the main header are situated close together. The same advantage can be obtained with the unit system if there are only three units. It will be seen that in Figs. 2 and 3, the valves *A A* are close together. Even when four units are installed the distance between the most remote valves need not be great. To show this, a fourth unit has been indicated by dotted lines in Fig. 2 and an additional valve A_1 provided.

It will be seen that in each of the arrangements shown in Figs. 1, 2 and 3, additions to the plant can be readily made without upsetting the design or involving complications.

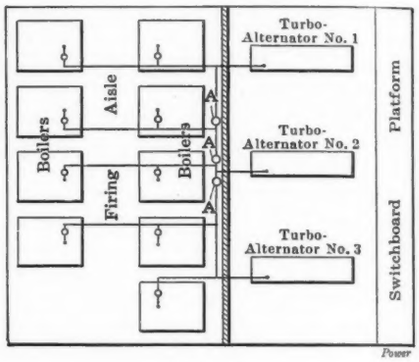


FIG. 3

R. M. NEILSON.

Glasgow, Scotland.

Trouble With Double Eccentric Engine

In a recent issue, E. T. Boynton brought before us a subject which should be better understood by the men in charge of engines of this type.

The engineer who understands only the ordinary slide valve knows that it would not do for the ports at both ends of the cylinder to be open when the valve is in the center of its travel, and if he were to open the valve chambers on a double-eccentric Corliss engine he would be very apt to look at the valves with the same eyes he had used on the slide valve and to think in a similar manner that the steam would rush right through without stopping, which it certainly does whenever the cutoff is not affected by the governor. It is well therefore on starting the engine to raise the governor block to such a position that the valves will trip. Of course, whenever the valves trip there is always one or the other of them closed. But when they "hang in the hooks" there is a time during every stroke when both of them will be open which results in an escape of steam into the receiver or the exhaust.

Two eccentrics are provided for the purpose of increasing the power of the engine by increasing the range of the

point of cutoff, for by means of two eccentrics it is possible to secure a cutoff at three-quarters stroke or maybe later, while with a single-eccentric Corliss engine the latest possible cutoff is about one-half stroke, assuming that the cutoff is affected by the governor and not by the return motion of the wristplate when the valves do not trip.

While it is all right for the valves of a single-eccentric engine to have lap when the wristplate stands central, it is always necessary to have negative lap with a double-eccentric engine, and when the wristplate is central and both valves are hooked up the steam ports at both ends of the cylinder should be from one-half to three-quarters open, because if they are lapped it will perhaps be impossible for the engine to deliver its rated load for the fact that the steam ports do not open wide, even with the latest possible cutoff.

I once took charge of two cross-compound Corliss engines rated at 750 horsepower and direct-connected to 500-kilowatt alternators. The manager told me that the engines had always proved unsatisfactory in that they would not carry their rated load. They had been purchased and accepted a number of years before when the load was light and there was no means of giving them a fair test. I found that if the steam pressure was kept right up to "popping off" point they would each carry from 400 to 450 kilowatts on a steady load but no more. The condensers had become unusable. I will not here attempt to tell why, for it would necessitate a long article, but it was principally due to the incompetency of the expert (?) who designed the plant.

The motion of one of the engines was uneven or pulsating, which at times set up cross currents between the generators when running in parallel; and the electrical engineer, under whom I worked, told me to see what I could do to make it run better. I put the high-pressure side on the inboard center and upon examining the wristplate found it to be about central also, so I loosened the steam eccentric and made the whole thing accurately central, made the links between the valve stems and wristplate of equal length, etc. I then removed the bonnets of the valve chambers and was nearly stricken dumb with astonishment for, unbelievable as it may seem, the head-end valve had a 7/8-inch opening, but the crank-end valve was lapped 1/8 inch. The engine had been running with about 1/4-inch lead at head end and a negative lead at crank end, which was the cause of the unevenness in speed. I drew the key from the bellcrank of the crank-end valve stem and turned the stem in the bellcrank until the valve corresponded with the other and made an offset key for the valve stem. Then I set the steam eccentric back until the lead came right and made a few small changes in the exhaust valves. After the engine was started again it ran like a different engine. The saving in coal resultant from this operation was from two to four tons per day for the same load and the engine will develop 700 kilowatts now as easily as it did 450 before.

Upon examining the valves of the other engine I found that both steam valves lapped when the valve gear was central. It was impossible to draw up enough on the links to the wristplate to get these valves where they belonged, so I made offset keys for both valve stems. I also had to make an offset key for one of the exhaust valves on the low-pressure side. After I got the valves of this engine to suit me it also easily handled 700 kilowatts.

C. F. CLARK.

Hartwick, N. Y.

Homemade Coal Testing Apparatus

In the article entitled "Homemade Coal Testing Apparatus," published in the March 15 issue of POWER, a correction is necessary in the following sentence: "For the smaller sizes of anthracite the following percentages of ash are the highest that should be permitted." This sentence should read: For the smaller sizes of anthracite the following percentages of slate are the highest that should be permitted.

W. P. ANDERSON.

New York City.

Boiler Leads

"For boiler leads, sizes are sometimes kept above the economical, due to the danger of priming with too high a velocity in the outlet. . . . . The best practice today is to allow a velocity of 4000 feet per minute in boiler leads when no special provisions exist against priming. This velocity is largely a matter of judgment, however, and experience must be relied upon in its determination." POWER, February 22.

What is a boiler lead? Is it the nozzle through which steam escapes from the boiler, or the pipe leading from that nozzle to the main steam pipe? The velocity of steam passing into the nozzle may possibly, when the water level is near to the nozzle, cause the boiler to prime, but what can the velocity in the pipe have to do with the priming if the pipe is bushed down from the nozzle, so long as the velocity at the nozzle is not enough to cause a lifting of the water?

Why does the "best practice" allow a velocity of 4000 feet per minute? Upon whose "judgment" is it based, and on what "experience"? Is it not a fact that the size of boiler leads, whether they are nozzles or pipes, has not been based on judgment or experience, but that, like Topsy, it just "growed"? How can anyone have a "judgment" as to what velocity in a lead will cause a boiler to prime when no one has any experience in the matter or has made any tests to find out what velocity will cause priming?

By the rule of 4000 feet per minute, a 4-inch pipe is large enough for a 121-horsepower boiler at 60 pounds pressure, and for a 212-horsepower boiler at 120 pounds, assuming 30 pounds per hour as a boiler horsepower. At 6000 feet per minute a 3-inch pipe will serve a 185-horsepower boiler at 120 pounds. Has anyone any experience from which he can form a judgment that 6000 feet is not just as good a figure to use for the velocity as 4000?

KENETH WILLIAMS.

Montclair, N. J.

A Broken Hook Block Bolt

I read with interest the article by A. W. Griswold in the February 8 number of POWER entitled, "A Broken Hook Block Bolt," and as I have had a little experience in the same line, I will submit it.

If the calenders and paper driers were being driven by the engine the best thing to do would have been to stop and make repairs, as these machines running with a fluctuating speed cannot turn out stock fit to be put on the market.

Mr. Griswold did not state whether the engine was running condensing or not, or if the cutoff of the low-pressure cylinder was automatically controlled. At any rate it would have been a good idea to have kept the receiver pressure lower than 50 pounds, especially if the receiver and low-pressure cylinder were of large diameters.

No doubt the engine referred to was running under full load, for otherwise the receiver pressure probably would not have gone up so high. The best thing to have done would have been to unhook the exhaust valve corresponding to the steam valve which was disabled and let enough steam blow through into the condenser or atmosphere, whichever the case was, to lower the receiver pressure to about 20 pounds. Then, I would have made the steam valve on the opposite end close as late as the eccentric would allow and I would also have made the cutoff later on one end of the high-pressure cylinder, depending on how the cranks were located, in order to keep the speed as steady as possible. In the meantime, I would have had the fireman raise the steam pressure a little. These changes could have been made while the engine was in operation if it were not traveling more than 120 revolutions per minute, and would have been all right in a place such as a large cotton mill where much time and money depend upon the continuous operation of the engine. If the reverse were the case

it would have been just as well to have stopped the engine at once and made the necessary repairs.

D. W. DUNBAR.

Attleboro, Mass.

Piston Movement of Duplex Pump

In the issue of January 11, A. G. asks which piston of a duplex pump will move first when the pistons are at opposite ends of the stroke. The answer given was correct but perhaps not as explicit as some might wish. This question is asked of applicants for license,

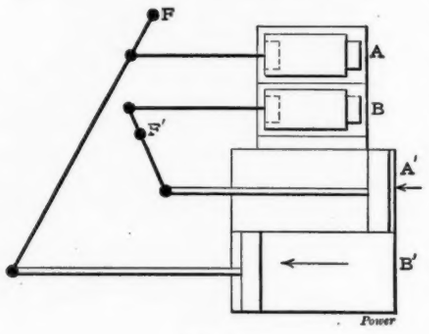


FIG. 1

and possibly to such persons a little further treatment of the subject will be welcome.

In Fig. 1, *A* and *B* are the valve chests, *A'* and *B'* the cylinders. The piston in *B'* moves the long lever whose fulcrum *F* is at the upper end. The valve rod is attached some distance below. The piston in *A'* moves the short lever whose fuclrum *F'* is between the piston rod and valve rod. The *A'* piston drives the *B* valve, and the *B'* piston drives the *A* valve.

Comparing this with a duplex pump, it will be seen that the long lever is con-

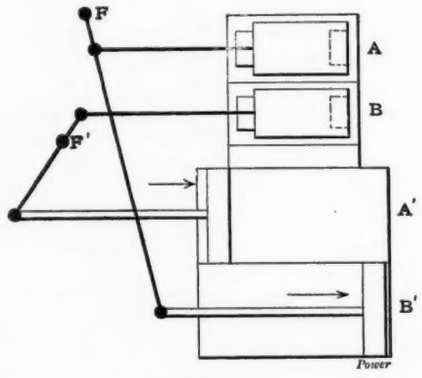


FIG. 2

nected to a bell crank reaching down and the short lever has a bell crank reaching up. This, then, is a correct representation. The long lever is a lever of the first class, and the short lever of second class. It is evident that the *B'* piston and the *A* valve will always move in the same direction; while the *A'* piston and the *B* valve will always move in opposite directions. With pistons as in Fig. 1, *A'* at head end, *B'* at crank end—both valves uncover head-end ports; *B* supplies steam to *B'* but that piston has reached the end of its travel; *A* supplies steam to *A'* whose piston is at the same end as the uncovered ports, and will therefore be the first to move.

In Fig. 2 the positions of the pistons are reversed, but it is again evident that the *A'* piston will move first. In either case the piston driving the short lever will move first. Similar diagrams made with pistons at same ends of stroke will show valves open at opposite ends and the piston driving the long lever will start first. If anyone is interested I advise him to draw such diagrams; also draw diagrams having both levers first class or both second class, showing that the pump would not work that way. This brings out the point of the matter which is, that while the long lever makes the valve follow, the short lever reverses its valve and makes continuous operation possible.

WILLIAM E. DIXON.

Hudson, Mass.

Kansas City Boiler Explosion

In the January 18 number of POWER there is a description of a boiler explosion which occurred in Kansas City on December 6, last. The heading of the article states that the probable cause of the disaster was a stop valve between the boiler and the safety valve. In my opinion, one other factor was present: low water.

I believe that had the boiler been sufficiently supplied with water, although it might have burst due to overpressure caused by the stop valve, it would not have exploded and caused a detonation like that of an artillery salute and wrecked the building to such an extent. My reason for believing this is based on an experience I had in 1852, when the boiler of the steamboat "Eastern City" burst and killed seven men by scalding.

The boiler was 14 feet in diameter and 34 feet long. The safety valve was set to blow at 40 pounds pressure. There were four gages of water in the boiler and the pressure gage indicated only 14 pounds at the time of the accident. It was discovered afterward that the gage was not connected into the boiler at all and the safety valve had become wedged. I stood 85 feet from the rear of the boiler when it let go. Hardly any noise was made at all. A great cloud of steam shot out in all directions and enveloped me as I ran up the gang plank. The bursting of the boiler caused no damage to the boat. It did not even disturb some loose planks which were standing on end against the wall near by.

PETER VAN BROCK.

Jefferson, Ia.

Ice Making Troubles

Reading an article in POWER of January 25, I was reminded of some experiences I have had along the same line as H. L. Vinton's.

I have been employed by the present company for six years and in that time have had considerable trouble with the quality of the ice, having made it in all the different shades of color from cherry red to jet black.

I cannot understand how Mr. Vinton eliminated red ice by cleaning the reboiler, cutting out the filters and repairing the leaky cooling coil. I consider his employer right in not supplying charcoal. The first season I was employed by the present company, the secretary was kept busy during one month writing letters of apology to our customers for having black ice for them at a time when ice was most needed, and a delay meant loss in time and customers.

I thought I would take a peep into the filters to find out where I was getting the red color from and sure enough I found it there, plenty of red and some oil. I concluded that new charcoal was what I wanted. The company purchased it without question, and the result was black ice and a loss of considerable money for the company. This was at a time when engineers were scarce so they did not give me the "grand bounce," but resolved to keep me until they could better themselves, and right at that time I commenced to better myself and have been improving ever since.

Instead of throwing the charcoal away every season, I learned that the makers had provided a system for cleaning it with steam when they had erected the plant, and I have been using it ever since with excellent results, except for eliminating the red; it stayed with me for two years. Cleaning filters did no good. The water before it was frozen looked transparent sometimes, while at other times it looked somewhat tinted.

The first conclusion was that the kind of oil I was using in the cylinder of the engine was to blame. I was always very careful about the excessive use of oil, and always blamed the night engineer for spoiling the entire twenty-four hours' output.

The company purchased the highest grade of cylinder oil they could get that was recommended for ice making. But the bad ice kept coming. The oil bill increased but brought no beneficial results.

Next, I turned to the reboiler, and without delay started to hunt for help. The manager was not so busy but that he wanted to go along. He at once decided that this was where the trouble lay and talked rather harshly about getting busy and delays costing money, etc., so we hurried the work to completion and started making ice again, and with

the same result—red ice. I had heard these two words so often that my hair which once was black was beginning to turn gray. And the boss—his hair began to stand up with anger. I began to realize that I was no longer a prophet, and that considerable time as well as money had been lost. I was worried almost to despair, and while in this weakened condition neglected to clean the lime and magnesia off of the steam condensers. I had always reserved this work for myself because it was at considerable distance away from the boss, and about all the quietness that I had was when I was there.

My condition was such that this work was left undone, and so marked was the improvement in the ice that the boss began to smile and asked what I had done. Of course, all I could say was nothing, which was the truth.

Having completed my sixth year the boss says he is going to keep me, not because of the good done in the past, but on account of the good I am going to do in the future.

J. F. HAMMERSLEY.

Newcomerstown, O.

Live Steam vs Exhaust Steam for Heating

Referring to George H. Gibson's article in the issue of February 22, I do not think he has gone quite far enough in his calculation of the amount of heat transmitted by a radiator supplied with superheated steam. His data are as follows:

In a 14-inch pipe in which superheated steam, at an average temperature of 590 degrees, was flowing at a rate of 16,000 pounds per hour, the heat loss by radiation was 0.854 B.t.u. per square foot per hour per degree difference between the temperatures of the steam and of the room.

When saturated steam at a temperature of 380 degrees was flowing in the pipe the coefficient of heat loss was 1.63 B.t.u. This latter figure corresponds closely to the figures commonly obtained in tests of cast-iron radiators with saturated steam at about 212 degrees.

Mr. Gibson calculates from these data that a radiator supplied with exhaust steam at atmospheric pressure will transmit about 54 per cent. more than it will transmit if supplied with superheated steam of the same pressure and a temperature of 247.5 degrees, obtained by passing saturated steam of 20 pounds pressure through a reducing valve, assuming in both cases a room temperature of 65 degrees; his calculation may be abridged as follows:

Superheated steam at 247.5 degrees; temperature difference equals

$$182.5 \times 0.854 = 155.9 \textit{ B.t.u.}$$

Saturated steam at 212 degrees; temperature difference equals

$$147 \times 1.63 = 239.6 \textit{ B.t.u.}$$

Increase equals

$$0.537 \times 155.9 = 83.7 \textit{ B.t.u.}$$

I submit that this calculation does not cover the case. It would be correct if the superheated steam flowed through the radiator so rapidly as to remain superheated and at an average temperature of 247.5 degrees; but it is far from being correct if the radiator is supplied, as is usual, with only enough steam to furnish the heat radiated. In this case the steam will rapidly be cooled and become wet steam, and the coefficient of transmission will become the same as that for exhaust-steam heating.

Let the radiator be a long pipe of 100 square feet of surface. Let x equal number of square feet passed over while the steam is cooling from 247.5 to 212 degrees, and $100 - x$ the number of square feet passed over while the steam is condensing at 212 degrees. Then mean temperature of superheated steam equals 230 degrees, temperature difference equals

$$230 - 65 = 165 \textit{ degrees.}$$

Heat transmitted per hour equals

$$x \times 165 \times 0.834 + (100 - x) \times 147 \times 1.63 = 23{,}961 - 380.5\,x.$$

As a first approximation to the value of x, assume the mean coefficient for the whole pipe to be 1.50 and the mean temperature difference 150 degrees, then the heat transmitted would be

$$100 \times 150 \times 1.5 = 22{,}500 \textit{ B.t.u.},$$

and from this x would equal 4 square feet. This amount of heat would be furnished by the condensation of

$$22{,}500 \div 970 = 23.2$$

pounds of steam at atmospheric pressure. Taking 0.48 as the specific heat of superheated steam this 23.2 pounds would give up heat in cooling from 247.5 to 212, or

$$23.2 \times 32.5 \times 0.48 = 352 \textit{ B.t.u.},$$

and this cooling would be effected by x square feet of cooling surface.

$$x = 352 \div (165 \times 0.854) = 2.5$$

square feet, nearly.

Taking as a nearer approximation, $x = 2.7$, we have B.t.u. transmitted equals,

$$2.7 \times 165 \times 0.854 + 97.3 \times 147 \times 1.63 = 23{,}694.$$

$$23{,}694 \div 970 = 24.43$$

pounds.

$$24.43 \div (32.5 \times 0.48) = 2.7$$

square feet.

The heat transmitted by the radiator supplied with superheated steam is 23,694 B.t.u., as compared with

$$100 \times 147 \times 1.63 = 23{,}961 \textit{ B.t.u.},$$

which would be transmitted by the radiator using exhaust steam. This is a difference of only 267 B.t.u., or a little over 1 per cent., instead of the 54 per cent. shown by Mr. Gibson's calculation.

WILLIAM KENT.

Montclair, N. J.

Weir Measurements

In criticizing an article by Mr. Van Winkle on "Weir Measurements," Mr. Abbott, in the February 15 issue of POWER, makes the two following statements, to which I take exception:

"The only way to gage the hight (of water on weirs) accurately is by means of the hook gage."

"I once gaged a stream by figuring the cross-sectional area at two different points 1000 feet apart, getting the velocity by timing floats with a watch and by taking the hight of the water with hook gages."

Mr. Abbott is quite correct in saying that a hook gage will give accurately the hight of the water on a weir, that is, within 1 or 2 per cent. But he is not correct in saying that a hook gage is the only accurate means.

A much more accurate method of finding the head on a weir is to use a water column constructed in a manner similar to a draft gage. If this is connected into the weir below the water level, the zero reading can be obtained by observing the hight of water in the column just as the water is about to flow over the weir and the final reading at whatever hight the water is, when the head is desired. The difference naturally gives the actual head. This method eliminates the corrections necessary when using a hook gage with appreciable velocities, and may be carried to almost any accuracy by decreasing the slope of the water column.

What I desire to criticize in the other statement is Mr. Abbott's method of getting the velocities by timing floats. What value are the results obtained, even if they check within 4 per cent. at two different stations, when the velocities used are altogether too high? Since the friction of the water against the bed of a stream causes that portion to move much slower than the surface water, the use of surface floats to find the velocity gives results considerably higher than the actual mean velocity. Constants are given to correct for this difference, though Mr. Abbott does not state that he made any correction, but the ratio of the mean velocity to the surface velocity is so variable, depending on the nature of the bed of the stream and the shape of the cross-section that at the best this method is extremely crude. An improvement on the surface-float method is to use a float connected to a weighted ball that can be adjusted to float at any desired depth. This will give the velocities at different depths and from these the mean velocity may be computed. While this method is a decided improvement over the one first given, it is subjected to rather large errors, and if results of a close degree of accuracy are desired, it will be necessary to use a current meter. In using a current meter it is customary to divide the stream to be measured into a number

of vertical sections and to get the velocity of the water at different depths in each section.

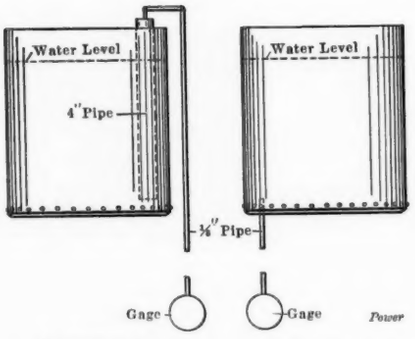
Anyone especially interested in the use of current meters is referred to an article in the August, 1909, number of the *Proceedings* of the American Society of Civil Engineers. Briefly, one of the most accurate and simplest methods is to take observations at 0.2 and 0.8 of the depth at the measuring point. The mean of the velocity at these depths is the mean horizontal velocity for the vertical section. In a similar manner by finding the velocity of each section and computing its cross-section the quantity of water for each section may be found and the sum of these quantities will give the volume of water flowing in the stream.

JOHN FRENCH.

Washington, D. C.

Water Tank Indicator

In a recent issue of POWER, a water-tank indicator was described by W. McAudell. His arrangement consists of a pipe in which air is entrapped and compressed so as to act on a gage calibrated to give the hight of the water in the tank. If this arrangement of Mr. McAudell's will work, why is it not a simpler arrangement to connect the gage directly to the tank and let the water into the gage? It is a well known fact that air is absorbed by water and it would be but a few months before the gage in the first case would have to be recalibrated; while as long as the gage will read accurately the hydrostatic head when connected directly, the hight of the water in the tank will be known. The gage



ARRANGEMENT SUGGESTED BY MR. MCAUDELL — SIMPLIFIED ARRANGEMENT

must be of a low-reading variety but still be of a large enough capacity to take care of the hydrostatic head between the gage and the bottom of the tank. If the gage is capable of considerable adjustment, this head can be taken care of in this way; and the zero point can be corrected to include this head. If such is not the case, a gage of such a capacity must be used that will show on the dial this head also. Roughly speaking, for every foot of head the gage will read about one-half pound per square inch. A small cock should be placed in the line near the gage, so that when a reading of the gage is made the cock can be opened for a moment beforehand. If water flows, it is a sign that the pipe to the tank is open. While it would be possible to calculate the hights of the water in the tank from the readings in pounds per square inch on the gage, it will be much easier to recalibrate the gage to the new scale. If the tank is not situated too high in the air, so as to make the hydrostatic head in the pipe to the gage considerable, this method should be a success.

G. A. GLICK.

Madison, Wis.

Simple Scale Remover

Our plant being small, I am the plumber, carpenter, engineer, fireman, and several other things.

We have a 60-horsepower boiler made by the Oil City Boiler Works. It is seven years old but looks as good as new inside. It gets a wash and thorough inspection every 30 days. Hearing so much about the use of kerosene in boilers, I thought I would write my little story which may be helpful to POWER readers.

I have used kerosene in our boiler with no bad effects but I know of something which seems to give better results. It is nothing more than a peck of potatoes just as they come from the grocery. The scale is broken up into particles like sand and in this form can easily be blown out occasionally.

URBAN L. EISEMAN.

Oil City, Penn.

Low Receiver Pressure

In the January 18 number, there is a letter by William Westerfield on "Low Receiver Pressure," which seems to need considerable explanation.

The incident was said to have happened in connection with the operation of a compound Corliss engine, the receiver of which had a gage having a range of 50 pounds. In his description Mr. Westerfield states that the receiver was subjected to a pressure of 140 pounds, which so distorted the spring of the gage that it afterward registered 20 pounds with no steam in the receiver.

He says: "When I arrived, I found the low-pressure cylinder was taking steam almost full stroke, and the receiver gage showed about 30 pounds pressure." He says that he applied the indicator and took diagram shown in Fig. 1. This diagram shows plainly that the receiver pressure was about five pounds below the atmospheric pressure, and I wish Mr. Westerfield would tell us how the receiver gage could show ten pounds above its new zero mark when there was a vacuum of about ten inches in the receiver?

Under the condition he states, the gage would actually have registered about 15 pounds instead of 30, as it would have been drawn back by the vacuum in the receiver.

Another peculiar point in his article is that the receiver gage, having a range of but 50 pounds, would have had to make practically two complete revolutions of its dial when subjected to the boiler pressure of 140 pounds, and it would be interesting to know just how the pointer managed to climb over the stop pin on the dial the number of times that it would have had to do so in going around and getting back, without being either bent, broken, or forced back on its shaft.

Possibly the engineer had the foresight to remove the pin from the dial before this interesting stunt took place. In explaining this, it is to be hoped that Mr. Westerfield will tell us what make of steam gage this was, and where the plant was located in which the incident took place. I believe it would be of interest to many readers if he would do so.

S. KIRLIN.

New York City.

Circulation Retarders

I read with interest Mr. Heglin's description in the issue of February 8 of an experiment made on a vertical water-tube boiler with a view to reducing the circulation by placing retarder plugs in the upper ends of the front tubes.

He states that these retarders had the effect of keeping a more solid body of water in the tubes. If the retarders had any effect whatever it must have been just the opposite. The amount of steam bubbles formed in the tubes depends on the quantity of heat transmitted from the fire to the water, and not on the area of outlet from the tube. Obviously the more constricted the outlet area is, the harder it will be for the steam bubbles to escape, and if they cannot escape they will stay in the tube and prevent a solid body of water from filling it. Evidently, therefore, the retarders could not have the effect of keeping a more solid body of water in the tubes, but would tend to prevent this.

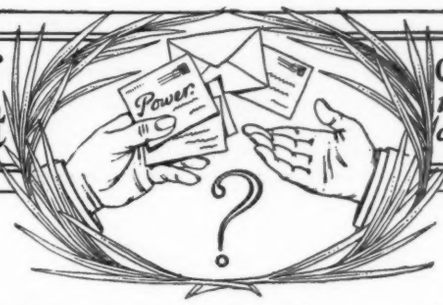
If Mr. Heglin had secured all the facts he would undoubtedly have found some other reason for greater tube losses occurring in those boilers not fitted with retarders than in the boilers with them.

CHARLES H. DELANY.

Elizabeth, N. J.

Up to February 1, 1696 Curtis turbines had been shipped, aggregating 1,301,286 kilowatts in capacity. Total orders up to February 1, 1910, were 1878 units, aggregating 1,524,208 kilowatts.

INQUIRIES of General Interest



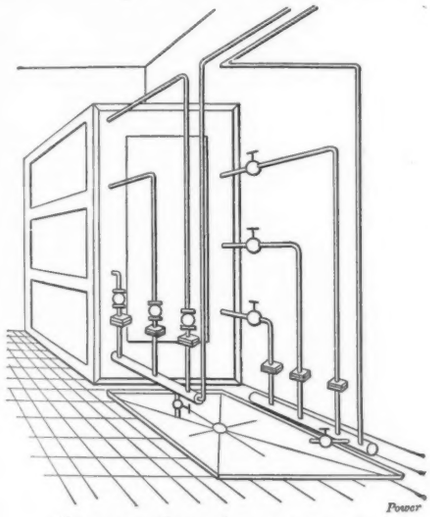
QUESTIONS are NOT ANSWERED UNLESS they are ACCOMPANIED by the NAME and ADDRESS of the INQUIRER.

Arrangement of Cooling Coils

Given an air cooler containing three coils of 1¼-inch pipe each 275 feet long, coils arranged longitudinally in three sections, as shown in sketch. Can more air be cooled by having the expanding ammonia vapor enter the coil at the lowest part and removing the expanded gas at the top instead of admitting it at the top and removing it at the bottom?

R. E. O.

So far as top and bottom expansion is concerned it is not believed that there will be any great difference in the two methods of passing the liquid except when the pipe surface cannot be used to the best advantage without danger of



COOLING COIL ARRANGEMENT

frosting back to the machine. When the expansion coils are short, the forcing of the coils is more apt to cause rushes of liquid to the compressor in the case of top than bottom expansion. If this action occasions the choking back of the expansion so that not all of the pipe surface is frosted thoroughly all of the time, there will be a loss of cooling capacity in proportion to the amount of surface and length of time such surface is not working efficiently.

The arrangement of the headers shown in the sketch should not cause trouble unless the coils are flooded sufficiently to allow the gas to carry over abnormal quantities of liquid.

If the liquid passes through the coils it will go to the machine, no matter what the piping arrangement may be. The interposition of liquid trap does little good as it eventually fills up and allows the gas to carry the entrained liquid to the machine in the same proportions it would had there been no trap installed.

If the several expansion coils are coated on the inside with oil, the horizontal arrangement is more disadvantageous than the vertical, helical, or "trombone" forms. This is because in the last-named cases the oil has more opportunity to flow toward the lowest point where the rapid boiling of the ammonia will have a better opportunity to carry it back to the machine with the return gas.

Without knowing more of the details of the system, the following suggestions are offered:

See that the expansion coils are free from oil on the inside and ice on the outside. See that the highest possible back pressure is carried. If forcing the coils sufficiently to freeze back to the return header makes the back pressure too high to cool the air, put in more baffles or weirs in the air chambers so that the air will have to come in contact with a greater amount of pipe surface before passing out of the cooling chamber. Put electric fan in the chamber temporarily and note whether the increased agitation of the air produces a drop in temperature for the same volume of air. If so, rearrange weirs so that the air has to travel a greater distance before leaving the chamber. Insulate the return headers if they are not run in cooled compartments. Insulate the liquid line if it runs through rooms materially warmer than your condensing water.

Superheating by Throttling

How does throttling steam superheat it, and is steam which has passed through a reducing valve hotter than it was before?

M. B.

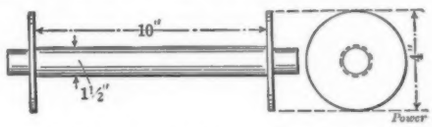
Consulting the steam tables it will be seen that a pound of saturated steam at any pressure contains more heat than a pound of saturated steam at a lower pressure. If steam at any pressure is passed through a reducing valve, except for the small amount lost in the work of forcing it through the valve, the steam will have the same amount of heat in it that it contained before throttling to the lower pressure. And as by definition superheated steam is steam which has a higher temperature than that due to its pressure the throttled steam is superheated although it may be slightly cooler than before throttling. It will never rise above the initial temperature.

Making a Permanent Bar Magnet

Please give instructions for making a permanent magnet of round steel 1¼ inches diameter and 12 inches long.

G. K.

Make a temporary spool for the magnetizing coil by wrapping a strip of paper 10 inches wide around the steel rod until you have increased the diameter to 1½ inches. With shellac varnish stick one convolution of paper to that already wound, at intervals of four or five layers. Make two heads for the spool, of ½-inch white pine or other soft wood, and put these on the rod 10 inches apart. Wind No. 30 single cotton-



BOBBIN FOR MAGNETIZING COIL

covered wire in the spool until the outer diameter is 4 inches. Connect the coil to a 110-volt direct-current circuit for a minute or so.

The steel must be hard, such as tungsten or chrome steel.

Absolute, Initial and Terminal Pressure

Will you please explain to a beginner in steam engineering what is meant by the terms absolute, initial and terminal pressure?

A. B.

Absolute pressure is the pressure measured from vacuum and is approximately 15 pounds per square inch above gage pressure, which is the pressure above the atmosphere.

Initial pressure is the pressure in the cylinder of an engine at or near the beginning of the stroke of the piston.

Terminal pressure is the pressure that would obtain in the cylinder of an engine at the end of the stroke if the exhaust valve did not open until then.

Area of Segment

On the inquiry page of the March 22 number, the formula for finding the area of a segment of a circle may be misleading due to the omission of a decimal point, making the 0.608 a whole number. The formula should read:

$$\frac{4\,H^2}{3}\sqrt{\frac{2\,R}{H}-0.608}$$

Power and The Engineer

Editorial

Priming

The present tendency is to burn more coal per square foot of heating surface, to get more evaporation per unit of boiler installation, to reduce the investment and other standing charges per horsepower by "increasing the denominator." Professor Perry has pointed to surprising possibilities in this direction, and recent articles in POWER by Messrs. Ray and Kreisinger have told of the findings of the United States Geological Survey in this regard.

There is no doubt that a lot more heat can be conveyed to water through a given amount of heating surface than is necessary to run a boiler at the usual rating of one-tenth of a horsepower per square foot of heating surface. To do so would mean less money in boilers and boiler house, less insurance inspection and all that goes to make up the standing charge against the boiler plant. The limiting factors just now are firebrick and priming, the difficulty of getting a furnace which will stand up and burn the larger amount of coal, and the difficulty of getting the greater quantity of steam separated out of the water without taking too much of the water along with it.

Aside from the danger which water in the steam involves in the way of broken engine cylinders, stripped turbine blades, broken steam mains, etc., priming involves a considerable expense. A 250-horsepower boiler furnishes 7500 pounds of steam per hour. Suppose this steam to be made from feed water of 200 degrees, to have 120 pounds (gage) pressure, and to contain 5 per cent. of moisture. This means that 375 pounds of water per hour have to be heated from 200 to over 340 degrees at an expense of 153.3 heat units per pound.

While this is considerably less than one per cent. of the total heat supplied, it amounts to something over ten tons of coal and 18,000 cubic feet of water in a 3000-hour year, which, especially where water costs metropolitan prices, is considerable of a price to pay for maintaining a nuisance in the steam pipes, imperiling cylinders and diminishing the efficiency of the engines. Separators returning the water to the boilers will correct a good deal of this, but the trouble is serious enough, especially in its practical effect of limiting the forcing capacity of boilers to warrant more attention in design and a study of the influence exerted upon it by various factors.

Auxiliary Fire Protection Systems

Few subjects have aroused greater interest among engineers than the recent discussions of water-supply and pumping systems designed to supplement or replace our time-honored methods of fighting fires. Their greatest usefulness is in the larger cities, where high values are concentrated in limited areas, and where the character and hight of the buildings, and the nature of their contents, add to the difficulties. This interest, however, is not confined to the larger cities but is extending to many smaller municipalities. That such systems are destined to even larger adoption may be accepted as a sign of the times.

The questions of hydraulics appeal primarily to the civil engineer, while those of drive—if electric—fall within the domain of the electrical engineer. Nevertheless the problem as a whole, and in most of its details, is clearly one for the mechanical engineer. It is common knowledge that he has risen to the occasion in characteristic fashion.

The requirements are many and severe. First in importance is reliability. The supply must be ample—the means for its delivery certain. There must be at least duplication of all parts, in most cases there is triplication, and in some cases even more. For, like the classic need of fire arms in Texas, when needed they are needed "bad." Reliability depends upon good design, fewness of units, simplicity of detail, ample strength of parts, and excellence of workmanship.

Next comes ability to get into operation quickly; then flexibility as to quantities of water and pressure. Low space occupied is important, as well as smallness of operating force. Last are low first cost, and low cost of operation and maintenance.

Is it any wonder that the engineer exults in the very joy of living when it falls to his lot to work out such problems as these:

Shall the motive power be steam, electricity, gas or oil?

If steam, what shall be the type of boiler, what fuel—hard or soft coal, gas or oil? What type of engine—reciprocating or turbine—condensing or not;

Issued Weekly by the
Hill Publishing Company
JOHN A. HILL, Pres. and Treas. ROB'T MCKEAN, Sec'y.
505 Pearl Street, New York.
355 Dearborn Street, Chicago.
6 Bouverie Street, London, E. C.
Unter den Linden 71—Berlin, N. W. 7.

Correspondence suitable for the columns of POWER solicited and paid for. Name and address of correspondents must be given—not necessarily for publication.

Subscription price $2 per year, in advance, to any post office in the United States or the possessions of the United States and Mexico. $3 to Canada. $4 to any other foreign country.

Pay no money to solicitors or agents unless they can show letters of authorization from this office.

Subscribers in Great Britain, Europe and the British Colonies in the Eastern Hemisphere may send their subscriptions to the London Office. Price 16 Shillings.

Entered as second class matter, April 2, 1908, at the post office at New York, N. Y., under the Act of Congress of March 3, 1879.

Cable address, "POWPUB," N. Y.
Business Telegraph Code.

CIRCULATION STATEMENT

During 1909 *we printed and circulated* 1,875,000 *copies of* POWER.

Our circulation for March, 1910, *was* 178,000.

April 5..... 36,000

None sent free regularly, no returns from news companies, no back numbers. Figures are live, net circulation.

Contents

PAGE

Plant of Pfister & Vogel Leather Company 614
Engine Driving by Compressed Air..... 618
Cooling Surface in Condensers..... 622
Flue Collapses in Lancashire Boiler..... 623
Precautions in Starting Steam Turbines 624
Water Power and Its Development..... 625
English Practice in Condensing Equipment 627
A Giant Motor..... 634
Simple Ground Detectors for Power and Lighting Circuits..... 635
Mr. Saeger's Problem in Storage Battery Connections 636
Possibly Excessive Line Drop..... 637
The Care of Rotary Converter Brushes.. 637
Splash Lubrication for Gas Engines..... 638
An Improved Water Pressure Gage..... 638
An Obscure Ignition Trouble..... 639
A Bad Case of Tar..... 639
Temporary Excess of Steam in a Producer 640
Performance of the Port Washington Gas Power Plant..... 640
Cooling Water Connections..... 640
Practical Letters:
Renewing Shaft Lining of Curtis Turbine .... Boiler Protection .... Governor Trouble .... Examination in Thermodynamics 641-643
Discussion Letters:
Steam Piping in Power Stations.... Trouble with Double Eccentric Engine .... Homemade Coal Testing Apparatus .... Boiler Leads .... A Broken Hook Block Bolt....Piston Movement of Duplex Pump....Kansas City Boiler Explosion....Ice Making Troubles....Live Steam vs. Exhaust Steam for Heating....Weir Measurements....Water Tank Indicator....Simple Scale Remover.... Low Receiver Pressure....Circulation Retarders.....644-648
Editorials 650-651

shall the steam be purchased or generated?

If electricity, shall it be direct or alternating, and at what voltage? If alternating, at what frequency and phase? Where may be found a supply ample in quantity, of absolute certainty, and at costs not prohibitive?

If gas, shall the supply be purchased or generated? What shall be the type of engine? Shall they be direct connected, geared or belted? If the gas is to be purchased what assurance is there of certainty of supply—and who is to provide the holders and mains?

If oil, are units of sufficient size to be had without excessive space requirements; are they reliable in every detail? Can the oil be had in sufficient quantities, at proper costs, and can it be so stored as not to itself add to the fire hazard?

Having reached some measure of decision as to these features, what of the pumps? Are they to be of the centrifugal or reciprocating type, vertical or horizontal, high or low speed?

All of these combinations are possible, in fact are in actual use, or in process of construction. There is as yet no accepted standard.

Another problem concerns the delivery mains; shall they be of cast or wrought iron? The former has heretofore been used almost exclusively on account of its durability, but it lacks reliability. Lines tested at high pressure fail afterward under much lower heads, without warning, and for reasons not yet fully explained. Wrought iron or steel, in spite of its admittedly higher depreciation, seems preferable on account of its reliability.

We violate no confidence in venturing the assertion that—taken by and large—no prettier problem exists today for the engineer. May the best plan win.

State Boiler Inspection

The unusually large number of boiler accidents which we have been compelled to record recently, attests the seriousness of the entire boiler question and emphasizes the need for closer inspection and more competent engineers in States where the law does not provide for such matters. In fact, there has never been a time in our industrial history when this question was of more importance than at present.

From time to time boilers are discovered operating under the most adverse circumstances and nearly every inspector can tell wonderful stories of boilers being run without safety valves or without some one of the other auxiliary devices that have come to be required as necessities by engineering experience. It is safe to say that at the present time there is not a State in the Union, aside from those in which laws covering this subject are in force, in which not one but many boilers are being operated in a positively dangerous condition, being kept from disintegrating apparently by an act of Providence alone.

The class of operatives in some of these plants must be improved if we are to expect better results in the future. The engineer of a Michigan plant not long ago solemnly declared to visitors that he was carrying 140 pounds of steam on two decrepit old "tea kettles," which statement so startled the aforesaid visitors that they made haste in the direction of the tall timber. It developed later, however, that this pressure was determined by adding the readings shown on each of the two steam gages, yet this man was the accredited chief engineer and had full charge of the plant. We have had occasion to see lately what this kind of operation means to life and property in Michigan, and most of the other States can duplicate it.

Another factor which undoubtedly has an influence on the increasing number of accidents is the desire to get the most work out of an already overloaded equipment. It seems to be the inevitable experience of all steam plants to have the load increase until it taxes the capacity of the machinery, and usually it will be found that factory engines are running loaded up close to their limit. Sooner or later comes the question of how to get more power out of the outfit. To increase the speed of the engine in the belt-driven factory will derange the adjustment of the machines, and to change the size of pulleys or flywheel will be expensive, and entail a shutdown. Neither is desirable, and obviously the quickest and easiest way is to increase the boiler pressure. Maintained for years satisfactorily at eighty pounds, we will say, the boiler pressure is run up to ninety or one hundred pounds. This, of course, has the effect of decreasing the factor of safety and, moreover, the decrease comes after a term of several years' service, when in the ordinary course of events the factor should be increased, which completely reverses the proper procedure from the standpoint of safety.

Increase in boiler pressure in such a plant should be made only after competent inspection and expert advice. Insurance companies can do much, but it is only after compulsory inspection laws are developed that we will be able to reach those horrible examples in the small obscure plant which, by the way, constitute in the aggregate the majority of the boiler horsepower installed in this country. By the term "small obscure plant" is not necessarily meant the isolated sawmill or rural manufacturing establishment. Many of these boilers are stuck away in the dark corners of highly populated localities. For instance, one of them, until a short time ago, was located under a sidewalk in Woodward avenue in the fashionable shopping district of Detroit. Compulsory inspection laws made it possible to seek out this boiler for examination, and the first tap of the inspector's hammer punctured the plate like cardboard. Uncounted thousands had passed daily over this subterranean mine for years without knowing anything of their danger.

This condition has passed in Detroit but will continue to be duplicated elsewhere until compulsory inspection in all the States comes to pass, combined with adequate license laws which will prevent the man who adds up his steam-gage readings from taking charge of a plant, and until then we can expect to see many repetitions of these boiler disasters in which the loss of life entailed will be absolutely to no purpose except as it helps in rousing public opinion to the importance of this subject.

The principal points to be observed in mechanical tests of lubricants are the effects of speed, load, temperature, and the frictional effects due to viscosity and oiliness, according to a paper presented to the American Society of Mechanical Engineers. The measurements on which depend the quality of the oil include the frictional resistance, the temperatures, and the endurance of the oil film.

Some of the experiences which have come to our notice recently indicate that the worst part of the failure of packing may be, not the lack of the packing at the place where it should be but its subsequent appearance at some point where it is a nuisance: stopping up valves, accumulating on strainers, etc. What becomes of all the old broken-down packing anyway?

Five million dollars a year is a conservative estimate of the damage done in Buffalo by soot from the chimneys of factories. This statement was made by M. C. Huyette in addressing the Buffalo Association No 16, N. A. S. E. Smokeless combustion is not altogether a matter of comfort or sentiment.

On March 10 a hearing was given at Albany, N. Y., on Assembly Bill No. 767, being an amendment to the education law providing for the licensing of civil engineers. The bill in its present form does not make it sufficiently clear that a civil engineer may be licensed to practise whether his education was obtained from a university or an institute of technology, or over the kitchen table after working in the field all day. Strengthened upon this point, so that it will provide for a certification of efficiency, however that efficiency was obtained, it will be an excellent measure.

NEW POWER HOUSE EQUIPMENT

WHAT *the* INVENTOR *and the* MANUFACTURER *are* DOING *to* SAVE TIME *and* MONEY *in the* ENGINE ROOM *and* POWER HOUSE

The Lytton Steam Trap

This steam trap, Fig. 1, is of a design peculiarly its own; designed by a practical engineer, familiar with the difficulties experienced in trap operation.

Fig. 2 shows the detail parts of the trap. It consists of a body made out of a piece of steel piping of suitable size and cut off at the desired length for the size of trap being made. By means of the through bolts shown, the two heads are clamped against the ends of this steel pipe body, the joint being made steam- and water-tight, on one end by a copper gasket and on the other by a vertical copper strainer, which also acts as a gasket.

The only moving part inside the trap is the bucket float, which is connected to the pipe *J*, as shown in Fig. 3. The valve is placed on the outside of the head, through which the pipe from the bucket float extends. The discharge valve is practically balanced, and is not dependent on the pressure in the trap for its movement; its size is not limited by the pressure and, therefore, a large valve opening is obtained with an increased discharging capacity. Fig. 4 shows an exterior, and Fig. 5 a sectional view of the discharge valve.

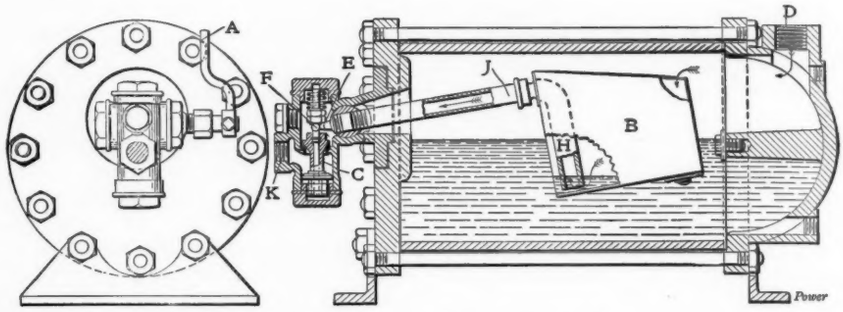


FIG. 3. SECTIONAL AND END VIEW OF TRAP

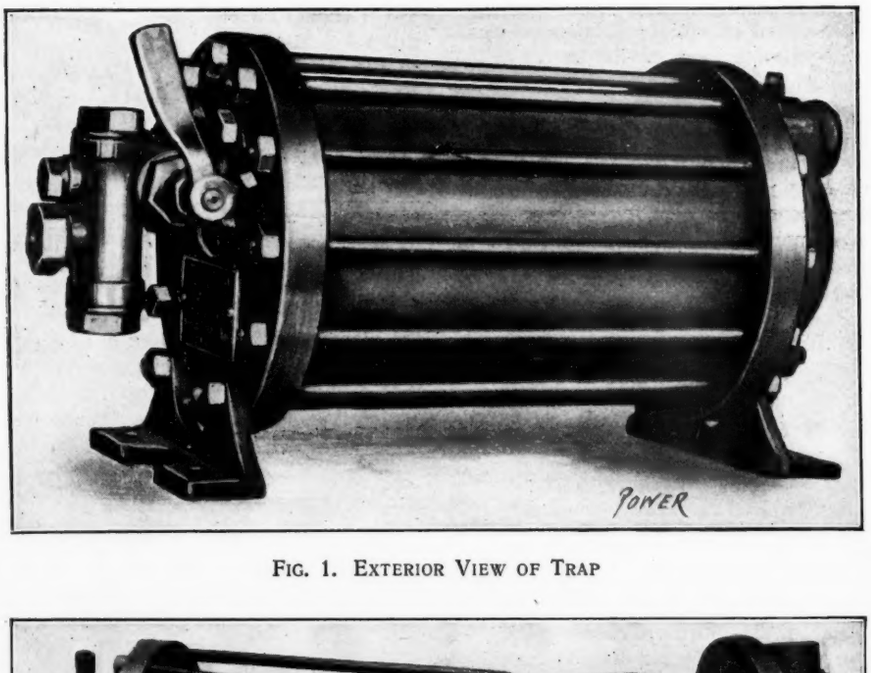


FIG. 1. EXTERIOR VIEW OF TRAP

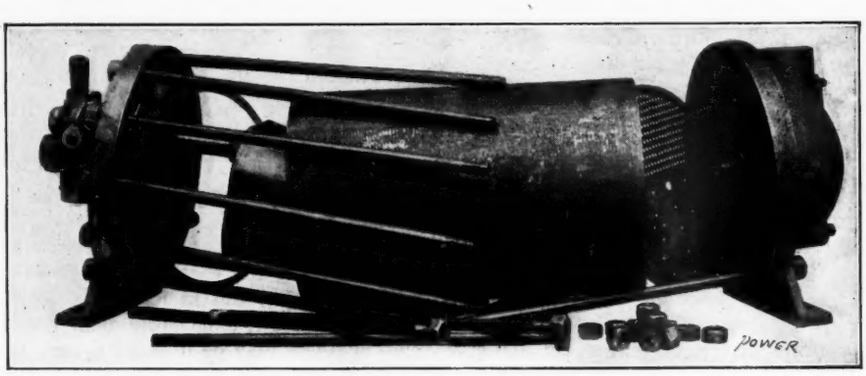


FIG. 2. TRAP DISMANTLED

A point which should not escape notice is the commendable feature of the feet or brackets, Fig. 2, which are bolted in place by means of two through bolts of sufficient length to accommodate the extra thickness of the detachable feet. This feature permits the placing of the trap on the floor, attaching it to the side wall, or suspending it from the ceiling, the only work necessary in changing positions being that of removing the feet and rebolting them to the trap heads in the position desired, so that the discharge valve will always stand in a vertical po-

FIG. 4

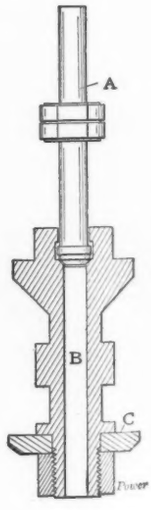


FIG. 5

sition. The operation of the trap is as follows:

Upon first installing the trap, it is necessary to either fill the receptacle approximately one-half full of water, or to hold the hand lever *A*, Fig. 3, forward, which will lift the float *B* and allow the discharge valve *C* to remain closed until sufficient condensation has flowed from the system to the trap to support the float in the position shown in Fig. 3.

When this condition has been established, the water flowing into the trap through the inlet *D*, will lift the float until it strikes the outer casing. When the trap has filled, the water will flow into the float *B*, and cause it to drop to the bottom. This movement produces an upward movement of the lever *E*, and lifts the auxiliary valve *F*, admitting the pressure in the trap to the under side of the piston attached to the main valve *C*. The main valve then opens and the water flows out of the trap through the float *B*, connection *H*, pipe *J*, and outlet *K*. The water continues to flow from the trap until it is level with the top of the float when at its lowest position. Sufficient water will then be blown from the float *B*, to permit it to rise to the position shown in Fig. 3. The last movement closes the discharge valve until the bucket again fills and drops to the bottom, when the operation is repeated.

The trap is designed to withstand a pressure of 150 pounds per square inch and the quantity of water that may be handled will vary directly with the given capacities according as the pressure may increase or decrease.

This trap is especially adapted for use on steam separators, steam pipes of engines and pumps, steam heating apparatus, etc., using either high- or low-pressure steam. The trap is manufactured by the Lytton Manufacturing Corporation, Franklin, Va.

Konkreto

This is a preparation for treating cement floors, walls and ceilings. It consists of a priming coat, which is first used on the cement surface, applying a little at a time until the pores and cracks will absorb no more. After an elapse of from 24 to 36 hours, this priming coat becomes set, the finishing coat is then applied, and about 24 hours allowed to elapse before the floor is ready for use.

This coating serves to prevent the surface of concrete floors from becoming ground into a fine powder by wear, gives a smooth and moisture-proof surface which is easily kept clean. One gallon of the priming and finishing preparation will cover from 300 to 500 square feet of surface, according to the porous nature of the concrete and the skill of the workmen. It is made in two colors, red and gray, by the Murphy Varnish Company, Newark, N. J.

A Fuel Economizer and Smoke Consumer

When water (H_2O) is heated to a sufficiently high temperature it is broken up into its component elements, hydrogen and oxygen. The Cornell patent fuel economizer and smoke consumer, manufactured by the Cornell Economizer Company, 55 De Long building, Philadelphia, Penn., is based on this fact.

In Fig. 1 is shown a return-tube boiler fitted with the Cornell system. Steam is taken from the steam dome and passed through a nest of retorts made of a special composition which is capable of withstanding an extremely high temperature. After becoming decomposed, the steam passes to air injectors which introduce it into the furnace both above and below the fire. The injectors are capable of being regulated so as to supply the exact amount of air needed to furnish the necessary oxygen to burn every particle of combustible.

The construction of the steam retorts is shown in Fig. 2. The steam passes up through the inner core and is forced down through the annular space between the core and the outside shell. The retorts are located in the hottest part of the path of the gases of combustion and are thus maintained at the white heat necessary to accomplish the decomposition of the steam.

Smoke consumption is accomplished by the upper sets of injectors which play directly across the top of the fire. Any smoke which may come off from the fuel comes into contact with this hot sheet of air and decomposed steam and is ignited. Without the interposition of this blast of hot gases the smoke would impinge on the comparatively cool plates or tubes of the boiler and its temperature would be so lowered that it would not ignite but would pass out of the chimney as wasteful and annoying smoke.

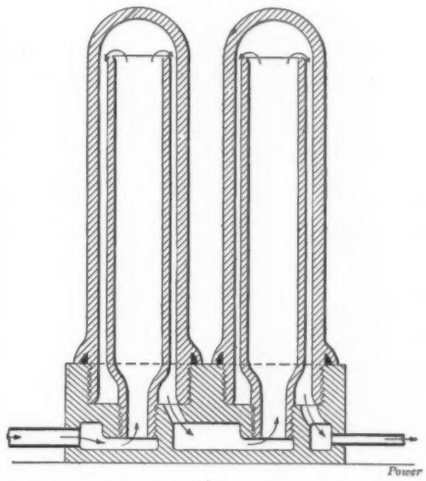


FIG. 2. LONGITUDINAL SECTION THROUGH STEAM RETORTS

The Cornell system is installed under a guarantee to save not less than 15 per cent. of the cost of the coal consumed, to prevent the formation of smoke from coal of any grade to the entire satisfaction of the local laws and ordinances and to remove the apparatus free of cost to the purchaser at the expiration of 30 days if the guaranteed efficiency has not been demonstrated.

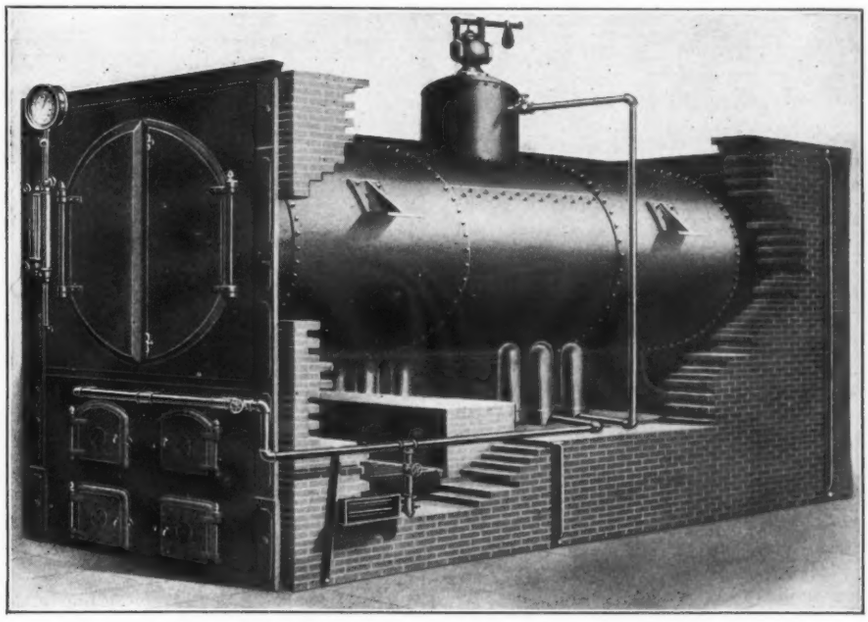


FIG. 1. TUBULAR BOILER EQUIPPED WITH CORNELL FUEL ECONOMIZER AND SMOKE PREVENTER

A Collapsible Steel Horse

A new collapsible steel horse has been placed on the market for the use of firemen, engineers, shop repair men and mechanics of various trades. It is made from steel angle iron, which allows of light but rigid and durable construction. Fig. 1 shows the features of design. The legs fold up when the horse is not in use, as shown in Fig. 2, making it convenient for storing away under a workbench or in the stock room. It does not take up much room, as shown by Fig. 3.

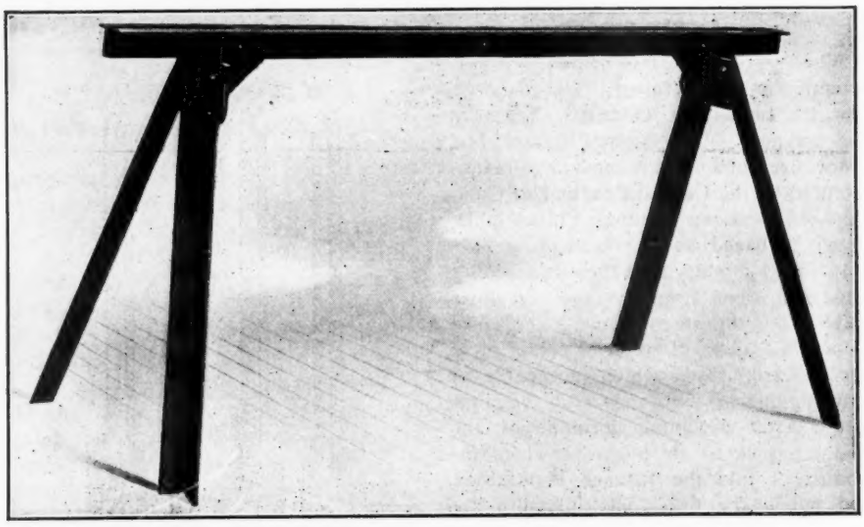


FIG. 1. COLLAPSIBLE STEEL HORSE

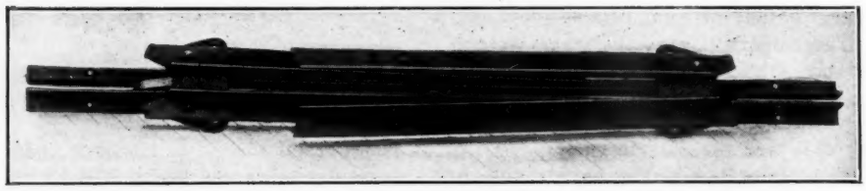


FIG. 2. HORSE FOLDED UP

The horse is made in six sizes, from 19 inches high by 20 inches long to 37 inches high by 35 inches long. Figs. 2 and 3 show a hole drilled in each end of the top piece for the purpose of securing boards or planks to the horses to form a table or work bench when desired. This is done by means of screws from the under side. The horse is prevented from folding when in use by pins placed through the holes in the frame.

By its use the old oil barrel, box or other makeshift, so often used in engine and boiler rooms for making repairs, blowing tubes, cleaning boilers, etc., can be discarded. This device is manufactured by the S. M. Hildreth Company, 2 Rector street, New York City.

In 1908 the United States produced over 415,000,000 tons of coal, a tonnage which exceeded considerably that of any country in Europe, the next being Great Britain with 292,000,000 tons of coal, with Germany a close third.

Return Steam Trap and Boiler Feed

A return steam trap unknown to many engineers, is manufactured by the American Machine and Manufacturing Company, Charlotte, N. C. This trap automatically returns the water of condensation from steam-heating systems or steam pipes directly back to the boiler. It consists of a cast-iron shell made as shown. The only interior movable part is a float attached to a stem, which is connected to a shaft running through a stuffing box. To this shaft a weight arm is secured, on which a weight is attached. The discharge valve is located on the top of the trap, and is operated by means of levers and links connected to the weight arm of the float.

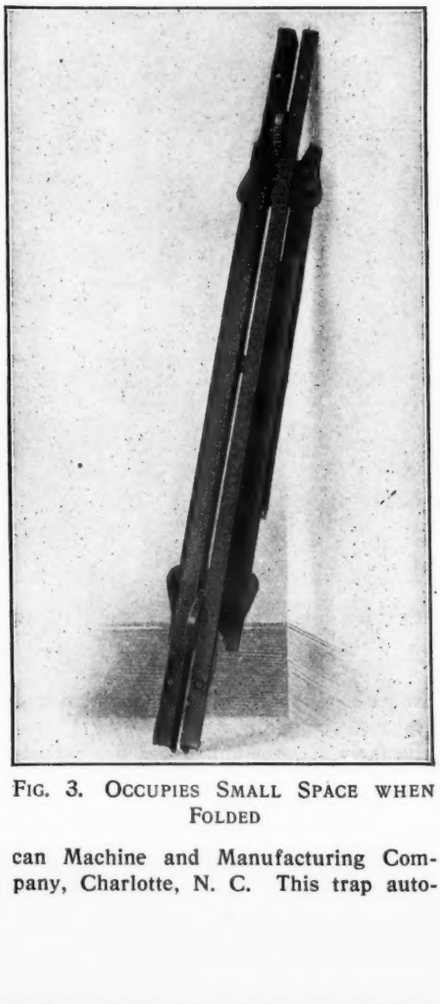
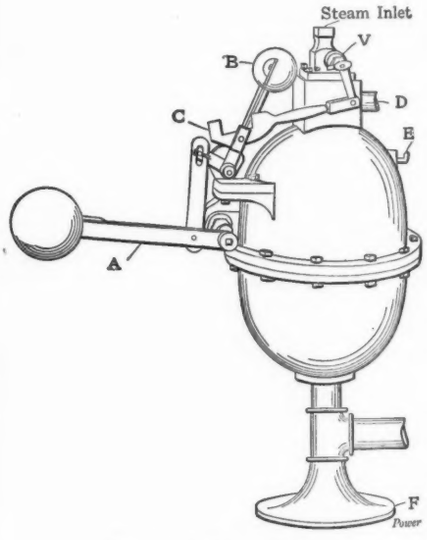


FIG. 3. OCCUPIES SMALL SPACE WHEN FOLDED

As the water in the trap raises the float, the weight lever *A*, descends, throwing the weight *B*, by means of the links, to a position opposite that shown, which forces the arm *C* with a quick movement toward the left. When this occurs, the pressure in the trap forces the water of condensation into the boiler. As the float descends, due to the change of water level in the trap, the weight *B*, assumes the position shown, thus forcing the lever *C* to the position illustrated, which shuts the steam-inlet valve. The exhaust outlet is on the opposite side, not shown, while the inlet for the condensation is at *D*. An air vent at *E* is for the purpose of permitting the air to escape while the trap is again filling with water of condensation. The discharge is through the bottom connection. A receiver is used in connection with this trap, to which the return pipes are connected.

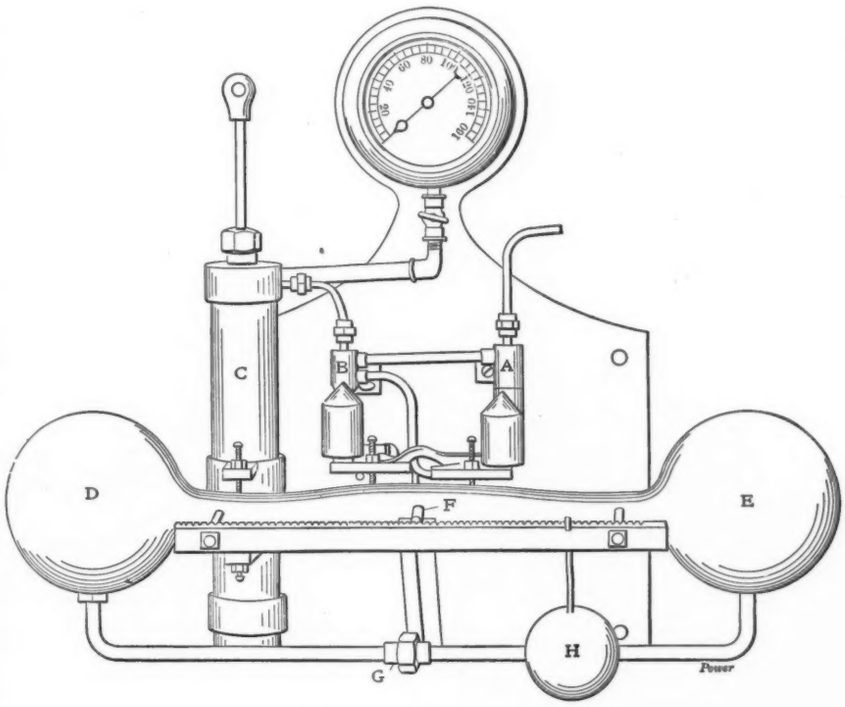


RETURN STEAM TRAP

In New England about 24,000,000 tons of coal are burned annually. Of the coal used in the industries for steam raising about 9,000,000 tons, or 58 per cent., are from Pennsylvania; 5,000,000 tons, or 32 per cent., from West Virginia; 1,000,000 tons, or 6 per cent., from Maryland, and 600,000 tons, or 4 per cent., from Nova Scotia. About 70 per cent. of all this coal is sent by water and about 30 per cent. by rail.

Economy Furnace Regulator

This regulator, manufactured by the Economy Valve Company, 336 The Arcade, Cleveland, Ohio, is illustrated and described herewith. It consists of three parts: the inlet and outlet valves *A* and *B*, the water cylinder *C*, and the mercury reservoirs *D* and *E*. These reservoirs are suspended on the knife-edge rest *F*, and the steam pressure is admitted to the reservoir *D* by a flexible copper wire.



ECONOMY REGULATOR

As the steam pressure increases, the mercury is driven through the bypass *G* to the reservoir *E*, which opens the inlet valve *A* and allows the water cylinder *C* to fill. This closes the damper by forcing the plunger in the water cylinder down, and also stops the feeding of the stoker. As the steam pressure decreases the mercury recedes through the bypass *G*, closing the inlet valve *A*, and opening the outlet valve *B*, which opens the damper and starts the stoker feeding.

It is claimed that any desired steam pressure, from 3 to 500 pounds per square inch may be carried by adjusting the weight *H*. The device is simple in construction and does not easily get out of order.

According to *The Providence Daily Journal*, the north plant of the Rhode Island Coal Company at Portsmouth was swept by fire on the afternoon of March 23. A spark from the stack serving the boiler plant is given as the cause of the fire, which resulted in a damage estimated at about $20,000. Besides the power equipment, the building in which the fire started also housed the supplies for the mine, and flying sparks soon set other buildings afire, including a blacksmith shop, the hoisting house and an oil house.

It is reported that very little was saved from the fire. A blower and forge, and a few tools, were taken out of the main building where the fire started. This structure was about 100 feet long by 40 feet wide. A quantity of oil was saved from the oil house before the fire reached that point.

Immediately after the fire, it is reported that workmen were engaged in clearing away the débris and were set to work on the foundations for a new boiler house, which had already been started. The company anticipated renewing operation within a few days.

PERSONAL

Walter M. McFarland, who has been associated with the Westinghouse Electric and Manufacturing Company since January 1, 1899, has resigned to accept an official position with the Babcock & Wilcox Company. Mr. McFarland has occupied the office of acting vice-president for the Westinghouse company for a period extending over ten years. In this capacity he has had official supervision of the large contracts of the company, as well as being the advisory head in all the coöperative movements of the company with the associated Westinghouse companies involving literature, advertising and exhibition work.

Mr. McFarland entered the United States Naval Academy as a cadet engineer in 1875, and in 1879 was graduated second in his class. In 1881 he was commissioned as assistant engineer; in 1891 as past assistant engineer, and in 1898 as chief engineer. In 1899, after the passage of the Personnel bill, he was commissioned lieutenant, and the same year resigned to enter the employ of the Westinghouse company.

After having had wide experience in sea service, he was detailed for service in the Bureau of Steam Engineering in 1882. From 1883 to 1885 he was detailed from the navy as assistant professor of mechanical engineering at Cornell University, and during the years 1885 and 1886 he was occupied with the inspection of machinery then building, and with work on preliminary design for proposed vessels. From 1889 to 1894 he was again attached to the Bureau of Steam Engineering.

This experience, together with his experience in the affairs of the Westinghouse company, admirably fits him for the duties which his connection with the Babcock & Wilcox Company will impose upon him.

WALTER M. MCFARLAND

Mr. McFarland was vice-president of the American Society of Mechanical Engineers in the year 1907, and at the present time is vice-president of the Society of Naval Architects and Marine Engineers. He is also a member of the Engineers Club of New York, Duquesne Club of Pittsburg, the Army and Navy Club of Washington, and the Army and Navy Club of New York.

Mr. McFarland will be located at the general offices of the Babcock & Wilcox Company in the Singer building, New York City.

OBITUARY

At six o'clock on the evening of March 19, Gardiner C. Sims, president of the William A. Harris Steam Engine Company, of Providence, R. I., died at his home, 272 Benefit street. Mr. Sims was in his sixty-fourth year. He had been ill for some time and became unconscious during the afternoon.

On July 31, 1845, Gardiner C. Sims was born at Niagara Falls. He received a common-school education in his native village, and as his inclination led to a trade rather than a profession, he obligated himself to an apprenticeship of four years to the machinist trade. This apprenticeship was served at the locomotive works of the New York Central & Hudson River Railroad Company, West Albany, N. Y.

THE LATE LIEUTENANT COMMANDER GARDINER C. SIMS

Three years were then spent as a draftsman in the engineering department of the United States navy yard at Brooklyn, and Mr. Sims then returned to the scene of his first labors as chief draftsman. Soon after, he accepted the position of superintendent of the J. C. Hoadley Engine Works, Lawrence, Mass., where he met Pardon Armington.

A partnership was formed, and the two men devoted their entire time to experimental work, which resulted in the development of their first high-speed engine. They built the first successful engine for Thomas A. Edison, which was sent to the Paris Electrical Exposition with his first dynamo in 1881.

In 1876 Mr. Sims spent eight months at the Centennial Exposition in Philadelphia, and was appointed Democratic commissioner from the State of Rhode Island to the World's Columbian Exposition. His prominence in the field secured for him the tender of the chairmanship of the exposition committee on electricity, electrical and pneumatic appliances, as well as membership on a committee of machinery and transportation.

In later years Mr. Sims had a notable career as an engineer in the Spanish-American war. At the outbreak of trouble with Spain he was quick to offer his services to the Navy Department and was appointed chief engineer, being ordered to the navy yard, Boston, Mass. One of his first duties was to fit up a repair ship for the fleet blockading the ports of the Spanish West Indies. The coastwise steamship "Chatham" was converted into a naval repair ship and was fitted out with machine tools, iron and brass foundry, boiler, blacksmith and coppersmith shops, also a supply of such tools and appliances as are found on warships. While his services were invaluable in getting the repair ship "Chatham" renamed the "Vulcan," ready for service, his activity in soliciting the services of machinists, molders, blacksmiths, coppersmiths, electricians and engineers to accompany him was noteworthy, the majority of those serving under him being acquaintances he had formed during the years he was engaged as engine manufacturer.

In recognition of his services he was made a lieutenant commander and received congratulatory letters from Secretary Long and Engineer-in-Chief George W. Melville.

In December, 1902, Mr. Sims was appointed police commissioner of Providence to fill the unexpired term of Richard H. Deming. For the last few years he has been connected with the William A. Harris Steam Engine Company, of Providence, as president. Mr. Sims leaves a widow and one son, who is a freshman at Brown university.

Mr. Sims was no poser and a satisfactory photograph of him has been difficult to obtain. For the full length, a snapshot taken during his later years, we are indebted to the *Providence Journal*. The larger portrait shows him as he looked when he and Pardon Armington answered the call of the electricians for a high-grade, high-speed engine.

NEW PUBLICATIONS

DIE HEISSDAMPF-SCHIFFSMASCHINE. (Superheated Steam Marine Engine). By Carl Fred Holmboe, Engineer. 60 pages, 6¾x9¾ inches, with 30 illustrations. Wilhelm Ernst & Sohn, Berlin. Price, in paper, 3.20 marks.

The subtitle characterizes this book as a collection of experimental data for calculating the dimensions and steam consumption, as well as the coal consumption, of marine engines for superheated-steam service. The results upon which it is based are partly the author's own, and in part are derived from other German and English practice. It is a brief handbook of design and thermodynamics in its special field. Appeal is made rather to the needs of the constructing engineer than to those of the running engineer unless he is prepared to enter quite seriously into the theoretical side of his occupation. In the first part we find formulas and diagrams relating to the principles of superheated steam, polytropic curves and computation of indicator diagrams, compression as a function of superheating, the Mollier heat diagram, etc. Part 2, beginning at page 18, leads into an inquiry concerning dimensions of cylinders, piping, etc., followed by steam consumption, mathematically treated. The third part is of more generally descriptive character and has to do with constructive details, piston valves, valve gears, stuffing boxes and lubrication. In the fourth and last part is discussed the economic significance of superheated steam for marine use. An appendix contains some brief steam tables.

NEW INVENTIONS

Printed copies of patents are furnished by the Patent Office at 5c. each. Address the Commissioner of Patents, Washington, D. C.

PRIME MOVERS

WATER MOTOR. Harry R. Irwin, Newport, Ky., and Henry C. Chambers, Cincinnati, Ohio; said Chambers assignor to said Irwin. 951,913.

ROTARY ENGINE. John F. Milton, Palestine, Tex., assignor of one-third to F. C. Bailey and one-third to C. J. Grainger, Palestine, Tex. 951,960.

ENGINE. James A. Dowling, Biloxi, Miss., assignor of one-fourth to Edward J. Dowling, Biloxi, Miss. 952,311.

INTERNAL COMBUSTION ENGINE. Louis F. Sleade, Denver, Colo., assignor of one-half to Willis A. Sleade, Silverton, Colo. 952,438.

BOILERS, FURNACES AND BURNERS

GAS BURNER. Thomas H. Mintier and Harry J. Jennings, Texarkana, Ark., assignors to Harry R. Segnar, Texarkana, Tex. 951,828.

HYDROCARBON BURNER. John L. Henry, Los Angeles, Cal., assignor of one-half to John C. Kenmonth, Los Angeles, Cal. 952,194.

BOILER SETTING. Edwin H. Cheney, Oak Park, Ill. 952,232.

LIQUID FUEL BURNER. Charles H. Speer, Chester, Penn., assignor of one-half to William M. Carr, New York, N. Y. 952,372.

FURNACE. Valentine H. Hobson, Richmond, Ky. 952,431.

ELECTRIC INVENTIONS

ELECTRIC SWITCH. Frank W. Smith, New York, N. Y. 951,851.

CARBURETER. Thomas H. Van Buren, Grand Junction, Colo., assignor of one-fourth to Cicero G. Smith, Palisade, Colo. 951,923.

ELECTRODE PROTECTOR. Herbert C. Harrison, Lockport, N. Y., assignor to Electro Metallurgical Company, a Corporation of West Virginia. 951,990.

ELECTRIC LAMP SOCKET. Johann G. Peterson, Hartford, Conn., assignor, by mesne assignments, to Marshall Electric Company, Hartford, Conn., a Corporation of Maine. 952,276.

SYSTEM OF ELECTRICAL DISTRIBUTION. Albert S. Hubbard, Belleville, N. J., assignor to Gould Storage Battery Company, a Corporation of New York. 952,332.

ELECTRIC IRON. Frederick Trott, Winnipeg, Manitoba, Canada. 952,385.

POWER PLANT AUXILIARIES, TOOLS AND APPLIANCES

PACKING GASKET. John W. Miller, Harrisburg, Penn. 952,128.

COOLING DEVICE FOR EXPLOSIVE ENGINE CYLINDERS. Francis M. Sizemore, Stewardson, Ill. 952,148.

VAPOR POWER PLANT. Charles P. Turner, Harrisburg, Penn. 952,386.

THROTTLE-OPERATING MECHANISM FOR EXPLOSIVE ENGINES. Alexander Winton and Harold B. Anderson, Cleveland, Ohio, assignors to the Winton Motor Carriage Company, Cleveland, Ohio. 952,398.

STARTING DEVICE FOR ENGINES. James T. Barber, Brooklyn, N. Y. 952,405.

AUTOMATIC PRESSURE-REDUCTION VALVE. Walter H. Bice, Maple Lodge, Ontario, Canada, assignor of one-half to Alfred Westman and one-sixth to William R. Colby, London, Canada. 952,410.

PRESSURE REDUCING AND REGULATING VALVE. Jehu V. Chase, Newport, R. I. 952,416.

METHOD OF OPERATING OIL ENGINES. Eugene Higgins, Jackson, Mich. 952,430.

ENGINEER'S VALVE. William H. Lieber, Milwaukee, Wis., assignor to Allis-Chalmers Company, Milwaukee, Wis., a Corporation of New Jersey. 952,434.

PACKING. George F. Miller, Easton, Penn. 951,827.

CONTROLLING MECHANISM FOR EXPLOSIVE ENGINES. Walter J. McVicker, Alma, Mich., assignor to Alma Manufacturing Company, Alma, Mich., a Corporation of Michigan. 951,833.

MECHANICAL PUSH BUTTON VALVE. Isaac G. Waterman, New York, N. Y. 951,862.

PIPE COUPLING. William H. Teuber, Carterville, Mo. 951,889.

WRENCH. George A. McIntire, Port Chester, N. Y., assignor to Perfection Wrench Company, a Corporation of New York. 952,079.

PUMP. John E. Gilmore, Sullivan, Ind., and William E. Gilmore, Winterhaven, Fla. 951,908.

VALVE GEAR FOR EXPLOSION ENGINES. Norman T. Harrington, Lansing, Mich. 951,910.

VALVE ACTUATING MECHANISM FOR GAS ENGINES. Valda W. Potter, Oelwein, Iowa. 951,965.

SPEED-CHANGING DEVICE FOR MOTORS. George W. Fowler and William P. Siebenmorgen, Westfield, N. J. 952,033.

VALVE. Eugene W. Hoyt, Decorah, Iowa. 952,043.

SURFACE CONDENSER. Charles G. Curtis, New York, N. Y., assignor, by mesne assignments to General Electric Company, a Corporation of New York. 952,102.

VALVE. James A. Donnelly, Brooklyn, N. Y. 952,187.

CARBURETER. Franklin W. Hagar, Nashville, Tenn. 952,326.

PUMP. Henry N. Murphey, East Oakland, Cal. 952,340.

PUMP. William H. Heard, London, Ontario, Canada. 948,302.

CLACK OR FLAP VALVE. Max F. Gutermuth, Darmstadt, Germany. 949,070.

WRENCH. Frank Mossberg, Attleboro, Mass., assignor to Frank Mossberg Company, a Corporation of Rhode Island. 949,083.

PIPE WRENCH. James F. Wright, Canton, Ohio. 949,107.

VALVE. Charles E. Davey, Detroit, Mich. 949,115.

ENGINE INDICATING APPARATUS. Frederick Purdy, Kenosha, Wis, assignor of one-half to Foree Bain, La Grange, Ill. 949,129.

CROSSHEAD. Baxter M. Aslakson, Salem, Ohio. 949,137.

IGNITION DEVICE. Baxter M. Aslakson, Salem, Ohio. 949,138.

AUTOMATIC CARBURETING MACHINE. Jacob F. Becker, Muskegon, Mich., assignor to U. S. Gas Machine Co., Muskegon, Mich., a Corporation of Michigan. 949,140.

VALVE. Thomas Andrews, Rockaway, N. J. 949,168.

MACHINE FOR FORMING ARTIFICIAL PRESSED FUEL. Ira Foreman and John L. Thornton, Los Angeles, Cal. 949,176.

VALVE. Charles R. Ballard, Midway, Penn. 949,208.

CENTRIFUGAL PUMP. Delevan V. Burrell, Rocky Ford, Colo. 949,215.

STEAM CONDENSER, WATER HEATER AND OIL SEPARATOR. John E. Caps, Chicago, Ill., assignor of one-half to Chapin A. Cartwright, Chicago, Ill. 949,217.

ROTARY PUMP. Otto E. Clement and John H. Wagner, St. Louis, Mo. 949,219.

REGULATING VALVE. George W. Collin, Bridgeport, Conn. 949,294.

PIPE COUPLING. Frank C. Pahlow, Detroit, Mich., assignor of one-half to Antonio C. Pessano, Detroit, Mich. 949,329.

SIGHT-FEED LUBRICATOR. Henry Weihman, Chicago, Ill., assignor of one-half to Edward Dieterla, Chicago, Ill. 949,339.

TIMER FOR INTERNAL COMBUSTION ENGINES. LeRoy C. Chowning, Corning, N. Y. 949,392.

PIPE WRENCH. Stephen A. Whetzel, South Brownsville, Penn. 949,557.

HOSE COUPLING. William J. Watt, Woodstock, Ontario, Canada. 949,608.

GASKET FOR PIPE JOINTS. Frank J. Randall, Los Angeles, Cal. 949,658.

PISTON RING. William A. Reinhardt, Ashland, Wis. 949,660.

WRENCH. Chauncey A. Rayman, Friedens, and Elmer Rayman, Coleman, Penn. 949,685.

WRENCH FOR SCREWING FLANGES OR SOCKETS ON PIPES. Martin McLane, Baltimore, Md. 949,711.

PACKING. David F. Stayman, Baltimore, Md. 950,519.

DRAFT REGULATOR. Henry B. Hemphill, Eldorado, Ill., assignor of one-fourth to Gein A. Sapp, and Carl E. Sapp, Wabash county, Ill. 950,918.

PIPE COUPLING. Charles Forth, Winthrop, Mass. 950,665.

RATCHET WRENCH. Samuel P. Perussina, Galveston, Texas. 950,675.

AUTOMATIC SWITCH FOR REVERSING GAS ENGINES. James M. Rhett, Beaufort, S. C. 950,704.

BOILER BLOW-OFF COCK. Jacob F. Zengles, Denver, Colo. 950,982.

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Pres., George Westinghouse; sec., Calvin W. Rice, Engineering building, 29 West 39th St., New York. Monthly meetings in New York City. Spring meeting at Atlantic City, May 31 to June 6.

NATIONAL ELECTRIC LIGHT ASSOCIATION

Pres., Frank W. Frueauff, Denver, Colo.; sec. and treas., Frank M. Tait. Association York. Next annual convention, St. Louis, Mo., May 23-28.

AMERICAN SOCIETY OF NAVAL ENGINEERS

Pres., Engineer-in-Chief Hutch I. Cone, U .S. N.; sec. and treas., Lieutenant Henry C. Dinger, U. S. N., Bureau of Steam Engineering, Navy Department, Washington, D. C.

AMERICAN BOILER MANUFACTURERS' ASSOCIATION

Pres., E. D. Meier, 11 Broadway, New York; sec., J. D. Farasey, cor. 37th St. and Erie Railway, Cleveland, O.

WESTERN SOCIETY OF ENGINEERS

Pres., J. W. Alvord; sec., J. H. Warder, 1735 Monadnock Block, Chicago, Ill.

ENGINEERS' SOCIETY OF WESTERN PENNSYLVANIA

Pres., E. K. Morse; sec., E. K. Hiles, 803 Fulton building, Pittsburg, Penn. Meetings 1st and 3d Tuesdays.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Pres., L. B. Stillwell; sec., Ralph W. Pope, 33 W. Thirty-ninth St., New York. Meetings monthly, excepting July and August.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

Pres., William G. Snow; sec., William M. Mackay, P. O. Box 1818, New York City.

NATIONAL ASSOCIATION OF STATIONARY ENGINEERS

Pres., William J. Reynolds, Hoboken, N. J.; sec., F. W. Raven, 325 Dearborn street, Chicago, Ill. Next convention, Rochester, N. Y., September, 1910.

UNIVERSAL CRAFTSMEN COUNCIL OF ENGINEERS

Grand Worthy Chief, W. S. Cadwell, Chicago, Ill.; sec., Thomas H. Jones, 244 Eighth street, N. E., Washington, D. C. Next convention, Buffalo, N. Y., August 2-5, 1910.

AMERICAN ORDER OF STEAM ENGINEERS

Supr. Chief Engr., Frederick Markoe, Philadelphia, Pa.; Supr. Cor. Engr., William S. Wetzler, 753 N. Forty-fourth St., Philadelphia, Pa. Next convention, Philadelphia, Pa., June, 1910.

NATIONAL MARINE ENGINEERS BENEFICIAL ASSOCIATIONS.

Pres., William F. Yates, New York, N. Y.; sec., George A. Grubb, 1040 Dakin street, Chicago, Ill. Next meeting, St. Louis, Mo., January 16-21, 1911.

OHIO SOCIETY OF MECHANICAL, ELECTRICAL AND STEAM ENGINEERS

Pres., O. F. Rabbe; sec. and treas., Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio. Next meeting, Cincinnati, May 19 and 20, 1910.

INTERNATIONAL MASTER BOILER MAKERS' ASSOCIATION

Pres., A. E. Brown; sec., Harry D. Vaught, 95 Liberty street, New York. Next meeting at Niagara Falls, Canada, May 24-27, 1910.

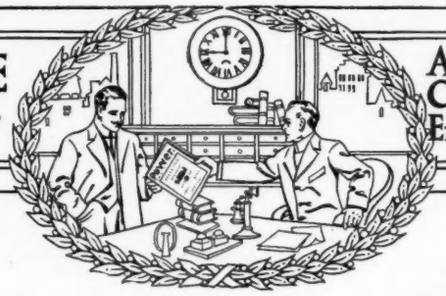
INTERNATIONAL UNION OF STEAM ENGINEERS

Pres., Matt. Comerford; sec., Robert A. McKee. Peoria, Ill. Next convention, Denver, Colo, September, 1910.

NATIONAL DISTRICT HEATING ASSOCIATION.

Pres., A. C. Rogers, Toledo, O.; sec. and treas., D. L. Gaskill, Greenville, O. Next annual meeting at Toledo, O., May, 1910.

MOMENTS WITH THE Advertising Editor



A Department for Subscribers Conducted by the Business End of Power and the Engineer

Once upon a time there was an Engineer who had charge of a power plant.

He managed to keep things running along all right most of the time, but nobody ever thought of pinning any medals on his vest.

The fireman called him an easy-going slob—the assistant called him happy-go-lucky—and the boss occasionally called him down.

He seemed to think he ought to let well enough alone, and contented himself with running an "average plant."

Now at home things were very different.

He used to sit around and talk most intelligently on horsepower and amperes to his wife who thought him a very clever cuss.

She didn't understand a word he said, but she knew her John was all the goods in his profession because he read his trade paper at night and kept right up with the red band wagon.

When any new device or improvement was advertised John studied it, and his comments were almost human at times.

In fact, his better ½ knew that he was an up-to-date engineer from his bald spot to his instep.

The thing she couldn't understand was why the boss of the plant didn't get wise to John's real value and add thickness to his pay envelope.

She didn't know what they thought of hubby in the engine room.

You see John really did know a whole lot, but he carefully hid the fact from everyone but his spouse.

So one day it came to pass that the Old Man sent for John to come to the office.

He was making a list of things from a copy of Power and The Engineer.

Said he, "John, look at these things—they are all new devices and appliances that are said to save much money in the power plant. What do you know about them?"

"Why," said John, "I know all about them. They are great savers of time, trouble and lucre."

"Then why haven t we got them in our plant?" asked the Boss. "Why haven't you told me of these things? Maybe they are new to you."

"Not at all," replied John, "why only last night I was telling my wife how much we need just those articles."

"But what in blazes has your wife got to do with it?" —the Old Man was angry—"She doesn't run this plant. I'm the one who pays the bills and you're the one I hire to keep them down. Here I've thought you were asleep at the switch when really you've known all along that these new devices are sadly needed. Now, you just make out a list of things and I'll send for them. I don't want to run a Louis XV power plant in the 20th century. And whenever you read about anything else that we ought to have, don't keep it to yourself. You know where I am."

And John went back to the engine room with a new idea.

He had found that there's something to do *after* you have read the ads and become posted—and that's to *take some action.*

You may find that some one or another device would save tremendously in your plant. You may read all about it in the advertising pages of the paper and you may get catalogs and data that give you a full understanding of its features and advantages.

But, unless you put it up to the Old Man to install it and try it out—all your knowledge is useless.

The moral is obvious:—Read the ads—study the articles advertised—*then act.*

An improvement in the plant is worth two in your mind.